AN EXPERIMENTAL STUDY OF ELECTRIC AND NATURAL GAS HEAT PUMPS

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Thomas Stephen Honeycheck

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AN EXPERIMENTAL STUDY OF ELECTRIC
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Approved:

Samuel V. Shelton, Chairman

Gene T. Colwell

Charles W. Gorton

Date approved by Chairman: 11/10/75

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SUMMARY

Commercial and residential space heating accounts for approximately one-third of the natural gas consumed in the United States, a larger percentage than any other single end-use. In order to better cope with present and future natural gas shortages, more efficient means of utilizing natural gas for space heating would appear to be an important area for investigation. From basic thermodynamics, a natural gas, heat-driven heat pump can be shown to have the highest potential efficiency possible. One configuration for consideration is a natural gas-fueled, rotary engine-driven, vapor-compression refrigeration cycle acting as a heat pump. Calculations for Atlanta show this system to have a significant improvement in efficiency over that of a natural gas furnace.

In this work an experimental analysis was conducted on a gas furnace, an electric heat pump, and a natural gas heat pump. A five ton (cooling) electrically-driven air conditioner was converted to a reversible heat pump by installing a four-way reversing valve. It was fully instrumented and its performance experimentally determined at various condenser and evaporator temperatures. The electrically-driven compressor was then replaced by an air-cooled, natural gas-fueled Wankel engine-driven compressor,

together with an exhaust heat recovery system. Similarly, for varying condenser and evaporator temperatures, this system's performance was determined. The results, as well as a comparison of the systems, are presented in this thesis.

Defining the COP for all systems as the desired output (cooling or heating) divided by the heat input to the system, the average COP_H of the gas furnace tested was .84. At a particular operating condition of T_L = 40°F and T_H = 120°F, the electric heat pump COP_C was .57 compared to the gas system's COP_C of .85. For the same temperature conditions, but in the heating mode, the electric heat pump COP_H was .75, while that of the gas heat pump with waste heat recovery was found to be .86. Better insulation of the exhaust system, thereby reducing the heat loss, however, could raise the latter value to 1.05.

CHAPTER I

INTRODUCTION

Background

An unavoidable consequence of the present energy crises is the need for more efficient utilization of our natural resources. Natural gas is a very important fuel for a number of reasons. Among the fossil fuels, it is the cleanest burning, requires no refining, and is easily recovered and shipped with little or no detrimental environmental impact. These factors all contribute to the justification of investigations of more efficient utilization methods of our natural gas resources.

In the United States, space heating accounts for more than one-third of the natural gas demand, a larger percentage than any other single use. Although the present day gas furnace has a rather high efficiency compared to other heating systems, basic thermodynamics shows that it can never approach the theoretical maximum heating efficiency, of which a natural gas heat pump is capable. Thus a natural gas heat pump might be one way to significantly improve the already high efficiency of natural gas heating and cooling.

Among the many possible types of natural gas heat pumps, one of the more promising is an engine-driven reverse

Rankine cycle. Unfortunately, poor engine reliability has been a major problem. Recent studies in the School of Mechanical Engineering at Georgia Tech, however, have shown that the natural gas-fueled rotary Wankel engine is capable of providing the necessary engine reliability [1].

Natural Gas Conservation

Fuel conservation is one of the prime advantages of a natural gas heat pump. Basic thermodynamics shows that such a heat pump can theoretically approach the ideal maximum heating efficiency that nature will allow. The national impact is quite impressive. Using pragmatic marketing assumptions, the resulting savings by 1990 would be 2.8 trillion cubic feet annually [1,2]. It is assumed in these calculations that the share of heating appliance market captured by the gas industry would continue to decline while the number of heat pump units was assumed to grow as in Table 1 [1,2].

Table 1. Heating Market Share (of Total Units Sold) [1,2]

Year	All Gas Systems	Heat Pump Units
1970	60.8%	
1976	52.3%	3.1% (140,000)
1980	51.2%	7.7% (762,000)
1985	50.2%	21% (1,853,000)
1990	50.0%	40% (3,416,000)

It should be kept in mind that the national coal gasification effort is expected to be producing three trillion cubic feet of synthetic gas annually by 1990 (slightly more than the 2.8 trillion cubic feet which could be conserved annually by the heat pump). The capital expenditure for coal gasification, however, is expected to be about $16 billion. In addition, the annual cost of the fossil fuel required by the gasification plant would be saved.

System Efficiency

In the analysis of any system an important parameter is the system efficiency, an indication of the system's merit. This term is a ratio of the desired effect relative to the input required to achieve it. In a power cycle, one is interested in the ratio of the net work output to heat input, which is termed the thermal efficiency. The concept of thermal efficiency, however, is not useful when applied to a reversed cycle (i.e. a refrigeration system or heat pump) because, in this case, the desired quantity is a transfer of heat, while the required input is some form of work or energy. Thus, an analogous term, called the coefficient of performance (COP), is usually considered. This term is of great importance in this thesis.

It would be wise at this point to consider the notation to be used throughout the following discussion.

The usual abbreviation (COP), which is used generally for coefficient of performance, can be modified by subscripts to COP_C or COP_H. The first form (COP_C) refers to a system in which the desired effect is a certain amount of heat removal (or cooling), while the latter (COP_H) refers to a system where heat addition (heating) is of prime interest. Whenever the forms COP_C and COP_H are used in this thesis, they refer to a cooling system in general and a heating system in general, respectively. When COP appears with no further notation, a general system efficiency is implied.

The concept of necessary input to the system suggests the need for further detail in the notation to be used. In this thesis, two types of COP's are defined depending on the form of input energy. In one form, the COP is defined as the cooling or heating effect relative to the net work input. The notation that will be used is COPW. Thus, $COPW_C$ and $COPW_H$ refer to the cooling and heating efficiencies based on work input to the system. The second case concerns the definition of COP as a cooling or heating effect relative to the energy content of the input fuel required to accomplish that transfer of heat. Thus, this COP is based on a heat input and the notation $COPH_C$ and $COPH_H$ will be used for the cooling and heating phases, respectively, of this type of system.

The use of the notation explained above, although perhaps somewhat different from common practice, was adopted to avoid problems arising from the use of COP in various contexts. The primary area in which clarification is required concerns the electric heat pump system. If the cooling or heating effect is divided by the electrical input rate, which is equivalent to a work input, $COPW_c$ or $COPW_H$ is the result. To obtain $COPH_c$ or $COPH_H$ for this same system the efficiency of producing the electricity must be considered. Throughout this work a power plant efficiency of 30% was assumed. Thus, for the electrical system, $COPH_c = (0.3)X(COPW_c)$ and $COPH_H = (0.3)X(COPW_H)$. This conversion is necessary because the COP's for the gas system are based on the natural gas input rate, a heat input. In order to compare the electrical and gas systems, common parameters are required and $COPH_c$ and $COPH_H$ can be used.

Heat Pump Gas Demand

The gas heat pump uses its fuel at a rather level demand rate, another important advantage. The monthly demand for a gas furnace ($COPH_H = .7$) is compared to that for the gas heat pump ($COPH_H = 1.3$, $COPH_c = 1.0$) in Figure 1 [1]. The amazing thing to realize is that the gas heat pump in Atlanta, Georgia, for example, would heat and cool year around with less total gas than a gas furnace would require for heating only. The furnace annual load factor (average monthly demand over peak demand) is 0.38 compared to the heat pump annual load factor of 0.63. These results were derived

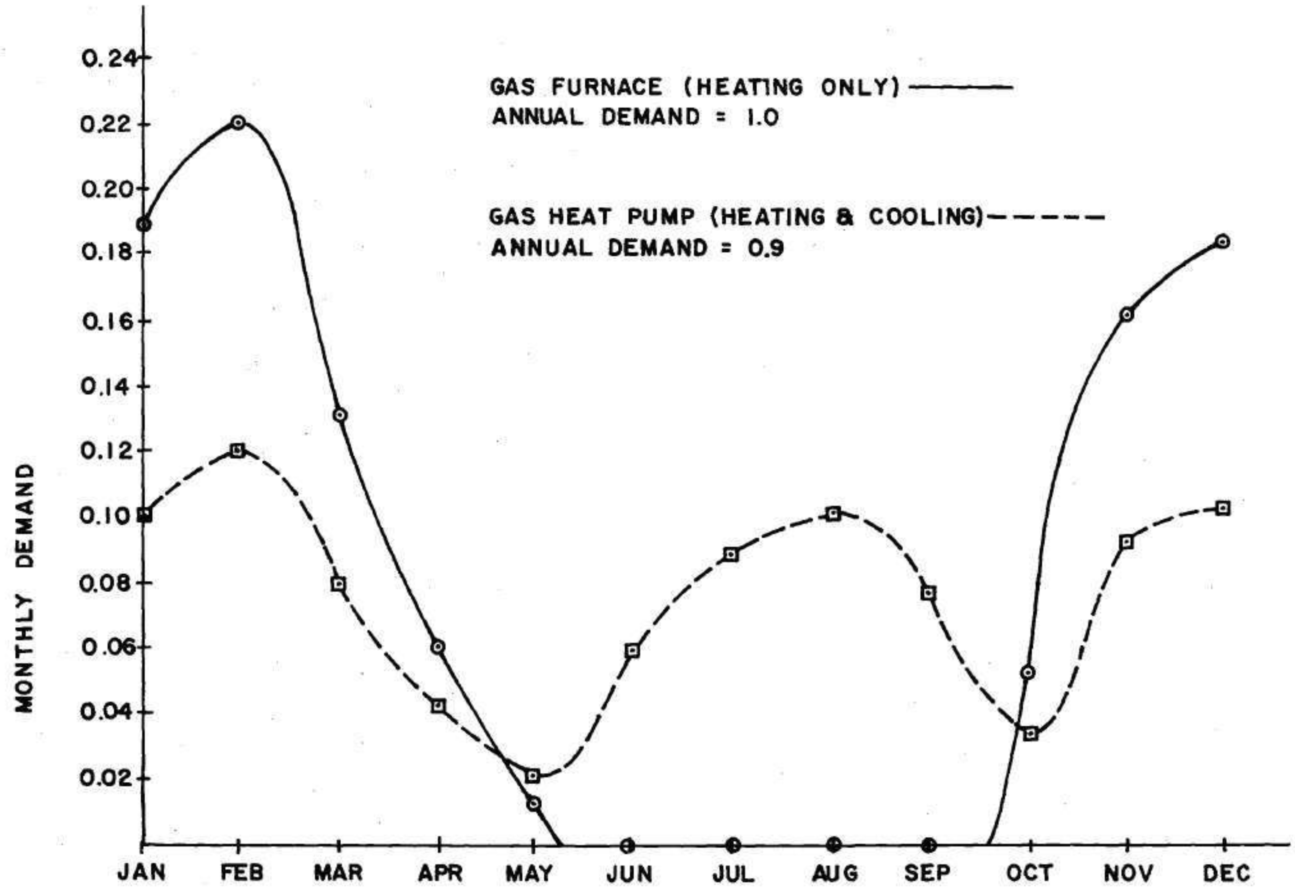


FIGURE 1.[1] MONTHLY GAS DEMAND FOR GAS FURNACE VS. GAS HEAT PUMP

for Atlanta, Georgia ten year weather conditions [1].

Air Pollution Impact

Environmental impact is an important area for consideration for any system proposed for widespread use. An electric resistance heating system with an overall system $COPH_H$ of 0.3, an electric heat pump with an overall system $COPH_H$ of 0.6, a gas furnace with a $COPH_H$ of 0.7, and a gas heat pump having a $COPH_H$ of 1.3 are compared in Tables 2 and 3 [1], assuming that the electricity for the electric systems is supplied by a 30% efficient coal-fired power plant burning 3% sulphur, bituminous coal, the type typically used in Georgia.

Table 2 compares the exhaust emissions based on the mass of pollutant emitted per 10^5 Btu of useful heat produced. Although the HC and CO emissions of the gas heat pump are higher than the other systems, a substantial reduction could possibly be achieved through the addition of a catalytic muffler on the engine. More important, however, is the gas heat pump's lower levels of NO_x and SO_x. Since the toxicity of SO_x is 125 times that of CO, two times that of HC, and 1.25 times that of NO_x, it would appear that the gas heat pump is better, healthwise, than the electric systems [1].

The total toxicity weighted emissions of each system are presented in Table 3. As can be seen, the gas heat pump emits three to five times less toxic material into the

Table 2. Emissions of Various Heating Systems (lbm pollutant)/(10^5 Btu useful) [1]

Method of Heating	Partic-ulates	NO_x	HC	CO	SO_x	Total
Resistance	.06	.16	.0023	.0088	1.0	1.231
Electric Heat Pump	.03	.08	.0012	.0044	0.5	0.615
Gas Furnace	0	.011	.0011	.0024	0	.0246
Gaseous Heat Pump	0	.056	.32	.097	0	.44

Table 3. Toxicity Weighted Emissions of Various Heating Systems (lbm pollutant)/(10^5 Btu useful) [1]

Method of Heating	Partic-ulates	NO_x	HC	CO	SO_x	Total
Toxicity Weighting Factor	1.06	.8	.5	.008	1.0	
Resistance	.064	.128	.0012	.000071	1.0	1.193
Electric Heat Pump	.032	.064	.0006	.000035	0.5	0.597
Gas Furnace	0	.0087	.00055	.000019	0	.0093
Gaseous Heat Pump	0	.045	.16	.00078	0	.21

atmosphere than the electric systems. Moreover, as explained in reference 1, the gas heat pump's toxicity weighted HC emissions are over estimated and the factor of two should be much lower. The overall conclusion to be drawn is that widespread use of a gas heat pump system would result in no detrimental impact on the environment, and would offer an environmental improvement over coal source electric heating.

Heat Pump System Concept

A heat pump is a system which transfers heat from a low temperature reservoir (T_L) to a higher temperature reservoir (T_H). Energy, usually in the form of rotating shaft work to a compressor, must be supplied to accomplish this transfer (see Figure 2).

A heat pump can be used to either cool or heat a room. During the summer, the low temperature reservoir which is giving up heat (being cooled) is the room air and the high temperature reservoir which is absorbing the rejected heat is the outside atmosphere. Thus room air conditioning is achieved. The common vapor-compression cycle, used to accomplish this, is shown in Figure 3.

If a natural gas-fueled engine is used to drive the compressor, instead of an electric motor, a natural gas driven air conditioning system results.

During the winter, heating may be accomplished simply by reversing the system. The indoor coil can serve

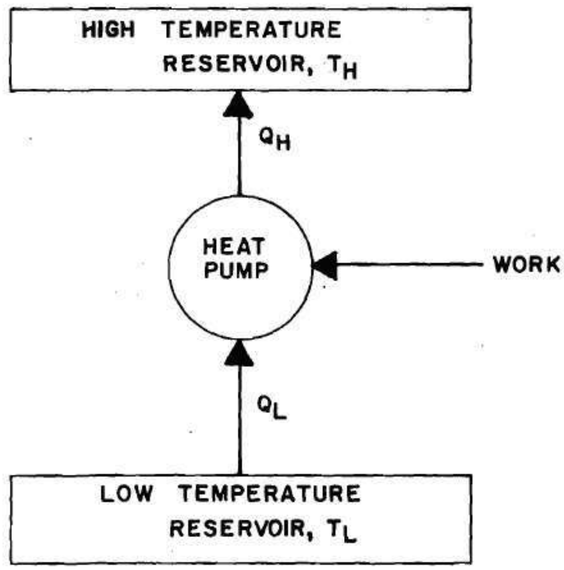


FIGURE 2.[1] THERMODYNAMIC HEAT PUMP

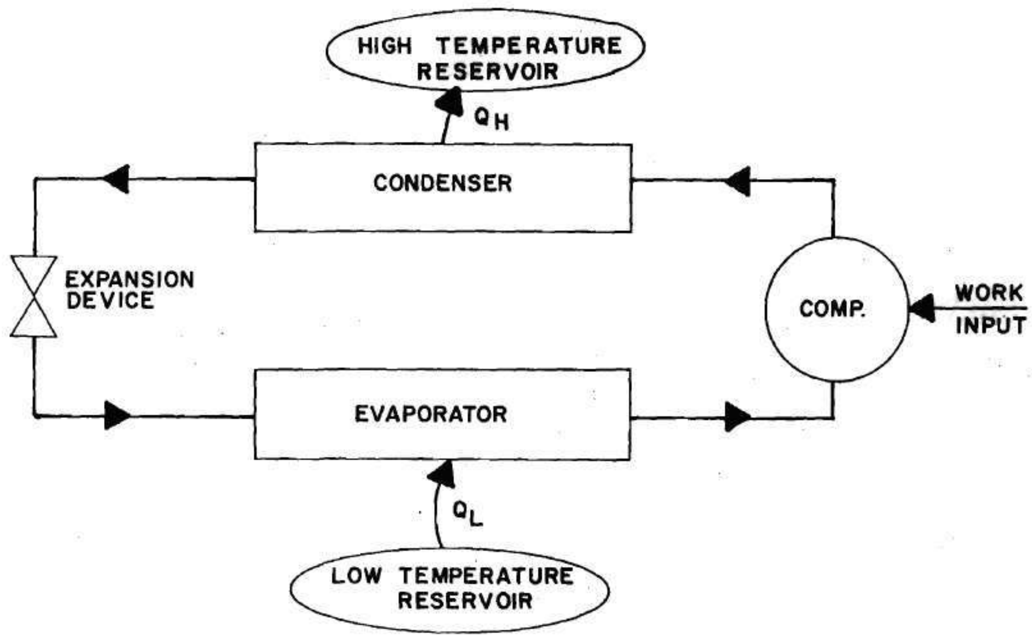


FIGURE 3.[1] VAPOR-COMPRESSION HEAT PUMP OPERATION

as the high temperature condenser to heat a room while the outside coil serves as the low temperature evaporator, absorbing heat from the atmosphere. Rather than having two systems, one for room heating and one for room cooling, a single system may be used and the condenser and evaporator functions interchanged simply by reversing the working fluid flow direction with a four-way reversing valve. If the work input to the system is provided by an electric motor, an electric heat pump results. If the work input is provided by a natural gas-fueled engine, a natural gas heat pump results.

The electric heat pump, by its basic nature, uses electricity more efficiently in space heating applications than does resistance heating. However, any electrical heating system has the disadvantage of having about two-thirds of the fossil fuel's combustion energy thrown away into rivers, lakes, or the atmosphere in the form of waste heat at the electric power plant. The power plant's remote location makes it too costly to utilize this waste heat resulting in thermal pollution as well as inefficiency.

In a natural gas heat pump, the waste heat, in this case in the form of hot exhaust gases, may be recovered to provide additional heat, as shown in Figure 4. This waste heat recovery results in a significant efficiency advantage over an electric heat pump. Thus, natural gas space heating is less costly.

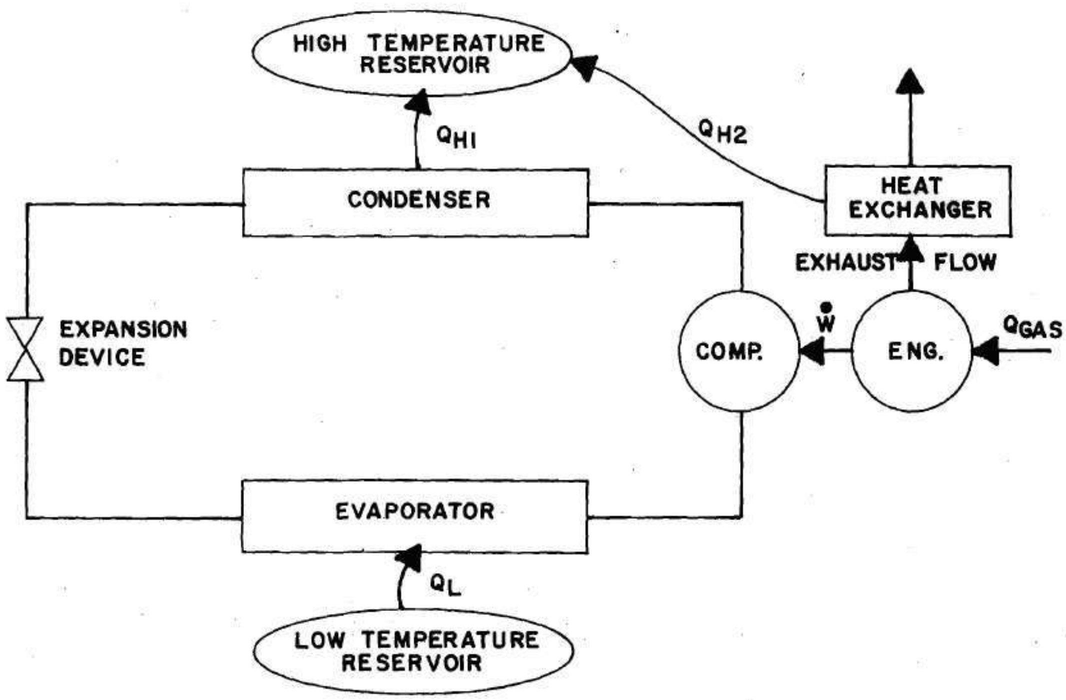


FIGURE 4.[1] NATURAL GAS HEATING BY A HEAT PUMP

Analogous to the comparison of the electric resistance furnace to the electric heat pump is the comparison of the natural gas furnace to the natural gas heat pump. The natural gas heat pump offers far more efficient utilization of natural gas than does a natural gas furnace due to the heat that the evaporator absorbs from the outside air.

This thesis will present for comparison the results of an experimental analysis of a natural gas furnace, an electric heat pump, and a natural gas heat pump.

CHAPTER II

HEAT PUMP PERFORMANCE ANALYSIS

Ideal Heat Pump Coefficient of Performance

The Clausius statement of the second law of thermodynamics is: "It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a cooler to a hotter body." The implication of this statement is that a system that does produce the transfer of heat from a cooler source to a hotter sink requires the input of some additional work or energy.

Basic thermodynamics show that the Carnot cycle for the working fluid of a heat engine produces the maximum work output for a given heat input. This simple cycle may be operated in the reverse direction resulting in a refrigerator or heat pump (see Figure 5). Heat is absorbed in this case from a low-temperature heat source, and a larger quantity of heat is eventually rejected to a high-temperature heat sink during the cycle. Work must be supplied from an external source to satisfy the second law. The difference between a refrigerator and a heat pump is mainly one of definition. The purpose of a refrigerator is to maintain a low-temperature source of finite size at a predetermined temperature by removing heat from it. A heat pump maintains a high-temperature

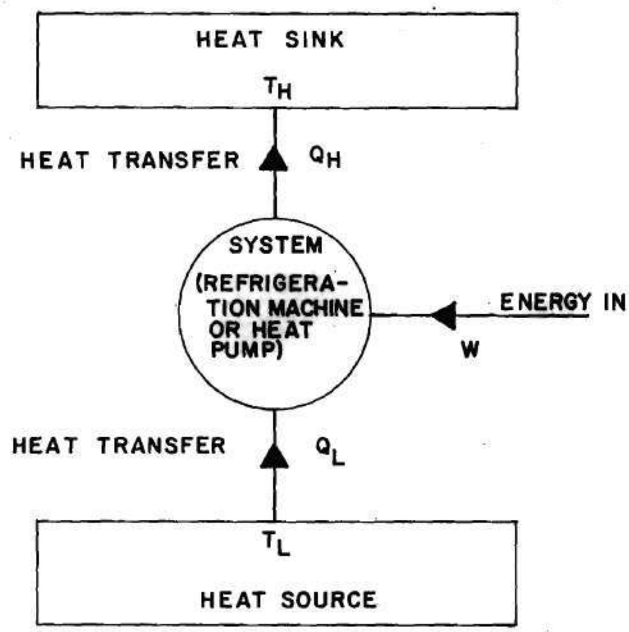


FIGURE 5. SCHEMATIC OF A CARNOT REFRIGERATOR OR HEAT PUMP

sink at a given level by supplying it with heat taken from a low temperature source.

When dealing with refrigerators and heat pumps, instead of thermal efficiency, the term usually desired is the coefficient of performance (COP). In refrigeration processes, the object is to remove the maximum quantity of heat per unit of net work input. Thus,

$$COPW_C = \frac{Q_L}{-W_{net}} = \frac{Q_{in}}{Q_{out}-Q_{in}} \tag{1}$$

The purpose of a shaft-driven heat pump is to supply the maximum quantity of heat to a high-temperature sink per unit of required net work input. Hence:

$$COPW_H = \frac{Q_H}{-W_{net}} = \frac{Q_{out}}{Q_{out}-Q_{in}} \tag{2}$$

The coefficient of performance for a Carnot, or reversible, cycle may be expressed in terms of temperatures as well as in terms of heat quantities. For heat pumps and refrigerators, heat is rejected to the high-temperature sink and heat is removed from a low-temperature source. Hence, the entropy change for a differential cycle driven by a reversible device is:

$$dS_{total} = dS_H + dS_L + dS_{device} = \frac{\delta Q_H}{T_H} - \frac{\delta Q_L}{T_L} + 0 \tag{3}$$

The maximum performance is attained when dS_{total} is zero. Thus:

$$\frac{\delta Q_H}{T_H} = \frac{\delta Q_L}{T_L} \quad \text{or} \quad \frac{Q_H}{Q_L} = \frac{T_H}{T_L} \tag{4}$$

Substituting equation (4) into equations (1) and (2) yields for a reversible work-driven system:

$$COPW_C(\text{ideal}) = \frac{Q_L}{W} = \frac{T_L}{T_H - T_L} = \frac{T_L/T_H}{1 - T_L/T_H} \tag{5}$$

$$COPW_H(\text{ideal}) = \frac{Q_H}{W} = \frac{T_H}{T_H - T_L} = \frac{1}{1 - T_L/T_H} =$$

$$\frac{T_L}{T_H - T_L} + 1.0 = COPW_C(\text{ideal}) + 1.0 \tag{6}$$

The above formulations are based on work input to the system. If one is concerned with a heat input instead, the equations are somewhat different. Consider a reversible heat engine operating between a high temperature source at T_S and a lower temperature sink at T_H, with the heat engine operating in the forward direction producing a certain amount of work (W). It can be shown that the thermal efficiency (η) of a Carnot or reversible heat engine is:

$$\eta = \frac{W}{Q_S} = 1 - \frac{T_H}{T_S} \tag{7}$$

or

$$W = Q_S \left(1 - \frac{T_H}{T_S}\right) \tag{8}$$

In these equations, Q_S represents the amount of heat input to the engine from the source at T_S. If this source represents the combustion process of an engine, for example, T_S would vary between approximately 3,000°F and ambient temperature. In this analysis, an average value of 1,000°F was assumed for T_S.

Consider a second engine operating between a source at temperature T_H and a sink at temperature T_L producing the same amount of work (W). Once again,

$$\eta = \frac{W}{Q_H} = 1 - \frac{T_L}{T_H} \tag{9}$$

or

$$W = Q_H \left(1 - \frac{T_L}{T_H}\right) \tag{10}$$

Consider now a combination of the two engines, the first a power cycle producing work (W), and the second a power cycle run in the reversed direction requiring an amount of work equivalent to (W). This combination produces a heat pump system in which the first engine provides the necessary input required by the second engine to accomplish a transfer of heat. Equations (9) and (10) are valid for a Carnot heat pump, because all quantities are the same as for a Carnot heat engine operating between the same two temperature levels. The only difference is that the directions of heat transfer are reversed and work is required rather than produced. The work is used to "pump" heat from the heat reservoir at the cooler temperature to the reservoir at the higher temperature. Since the two work terms must be equal, equations (8) and (10) can be equated:

$$Q_s \left(1 - \frac{T_H}{T_s}\right) = W = Q_H \left(1 - \frac{T_L}{T_H}\right) \tag{11}$$

The overall system considered here can be used to provide either cooling or heating. In the cooling mode, the system is using a quantity of heat (Q_s) from the high temperature source to remove a quantity of heat (Q_L) from a low temperature reservoir, while rejecting a quantity of heat equal to (Q_H). An energy balance indicates that:

$$Q_s + Q_L = Q_H \tag{12}$$

Substituting equation (12) into equation (11) yields:

$$Q_s \left(1 - \frac{T_H}{T_s}\right) = Q_s \left(1 - \frac{T_L}{T_H}\right) + Q_L \left(1 - \frac{T_L}{T_H}\right) \tag{13}$$

or

$$\frac{Q_L}{Q_s} = \frac{T_L/T_H - T_H/T_s}{1 - T_L/T_H}$$

Thus, a COP based on heat input to the system can be defined for the cooling phase as follows:

$$COPH_c(\text{ideal}) = \frac{Q_L}{Q_s} = \frac{T_L/T_H - T_H/T_s}{1 - T_L/T_H} \tag{14}$$

In the heating mode, the desired quantity is the heat rejected to the high temperature reservoir (Q_H). Recall:

$$Q_s \ (1 - \frac{T_H}{T_s}) = W = Q_H \ (1 - \frac{T_L}{T_H}) \tag{11}$$

For a heat pump system with heat as the input,

$$COPH_H(\text{ideal}) = \frac{Q_H}{Q_s} = \frac{1-T_H/T_s}{1-T_L/T_H} \tag{15}$$

The Vapor-Compression Cycle

The Carnot cycle cannot be achieved by a real working fluid. In practice, it is approached best by the vapor-compression cycle, the most important refrigeration cycle from the standpoint of commercial acceptance. The relatively high latent heats of vaporization and condensation for most fluids allow large refrigeration effects for modest rates of circulation.

The standard vapor-compression cycle is shown schematically in Figure 6, on a T-S diagram in Figure 7, and on a P-h diagram in Figure 8. The processes which comprise the standard vapor-compression cycle are:

1-2: reversible, adiabatic compression from saturated vapor to superheated vapor at the condenser pressure

2-3: heat rejection at constant pressure, desuperheating and condensation

3-4: irreversible expansion at constant enthalpy from saturated liquid to a liquid-vapor mixture at the evaporator pressure

4-1: heat addition at constant pressure in evaporation to saturated vapor.

The only difference between the heating and cooling modes is the position of the evaporator and condenser.

With the aid of a pressure-enthalpy diagram, Figure 8, the significant quantities of the cycle can be easily determined. Among these are the work of compression, the heat rejection, the refrigerating effect, and ultimately, the coefficient of performance.

The work of compression in Btu/pound of refrigerant is equal to the enthalpy change in process 1-2. From the steady-flow energy equation, with negligible changes in kinetic and potential energy, the work (W) in an adiabatic compression is simply:

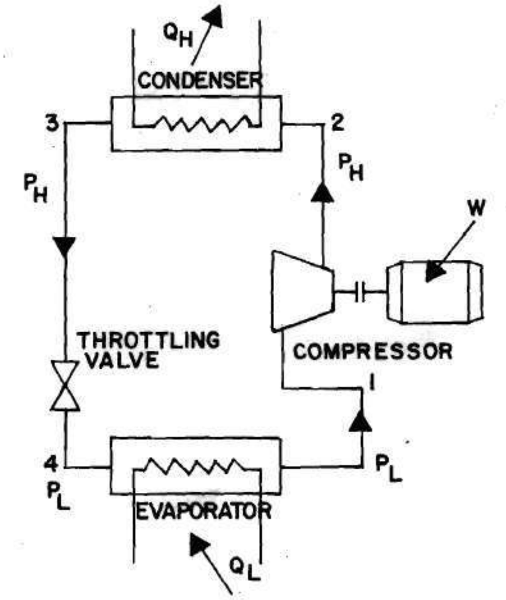


FIGURE 6. THE STANDARD VAPOR-COMPRESSION CYCLE

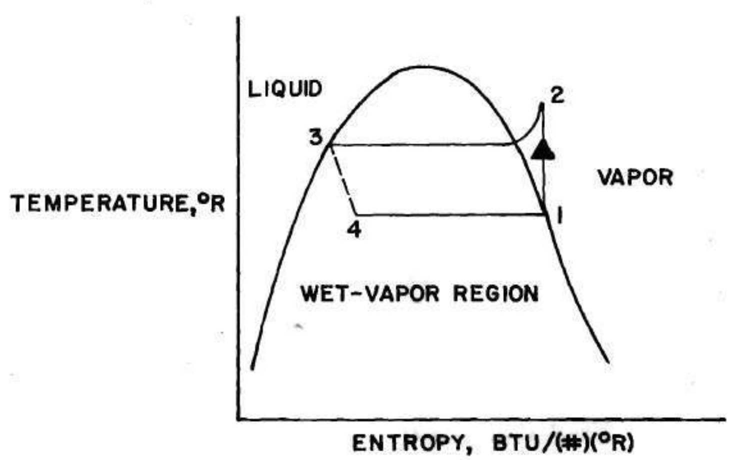


FIGURE 7. THE STANDARD VAPOR-COMPRESSION CYCLE

$$W = h_1 - h_2$$

The enthalpy difference is a negative quantity, indicating that work is done on the system. A knowledge of this quantity is important since it is the largest operating cost of the system.

The refrigerating or cooling effect in Btu/pound is the heat transferred in process 4-1, or $h_1 - h_4$. Since this process is the ultimate purpose of a cooling system, any further explanation of its importance is unnecessary.

The heat rejection in Btu/pound is the heat transferred from the refrigerant in process 2-3, which is $h_3 - h_2$. This relation also comes from the steady-flow energy equation in which the kinetic energy, potential energy, and work terms drop out. The value of $h_3 - h_2$ is negative, indicating that heat is transferred from the refrigerant. In the heating cycle this quantity is the desired effect.

The actual vapor-compression cycle suffers some inefficiencies compared with the standard cycle, as well as other changes that intentionally or unavoidably occur. Superimposing the actual cycle on the P-h diagram of the standard cycle, as in Figure 9, may be useful for comparison.

The essential differences between the actual and the standard cycle appear in the pressure drops in the evaporator and condenser, in the subcooling of the liquid leaving the condenser, and in the superheating of the vapor leaving the

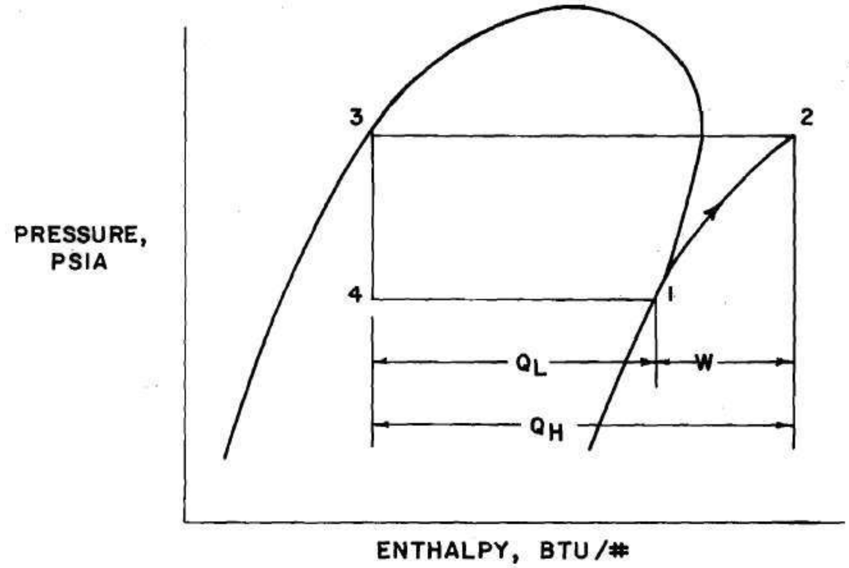


FIGURE 8. THE STANDARD VAPOR-COMPRESSION CYCLE

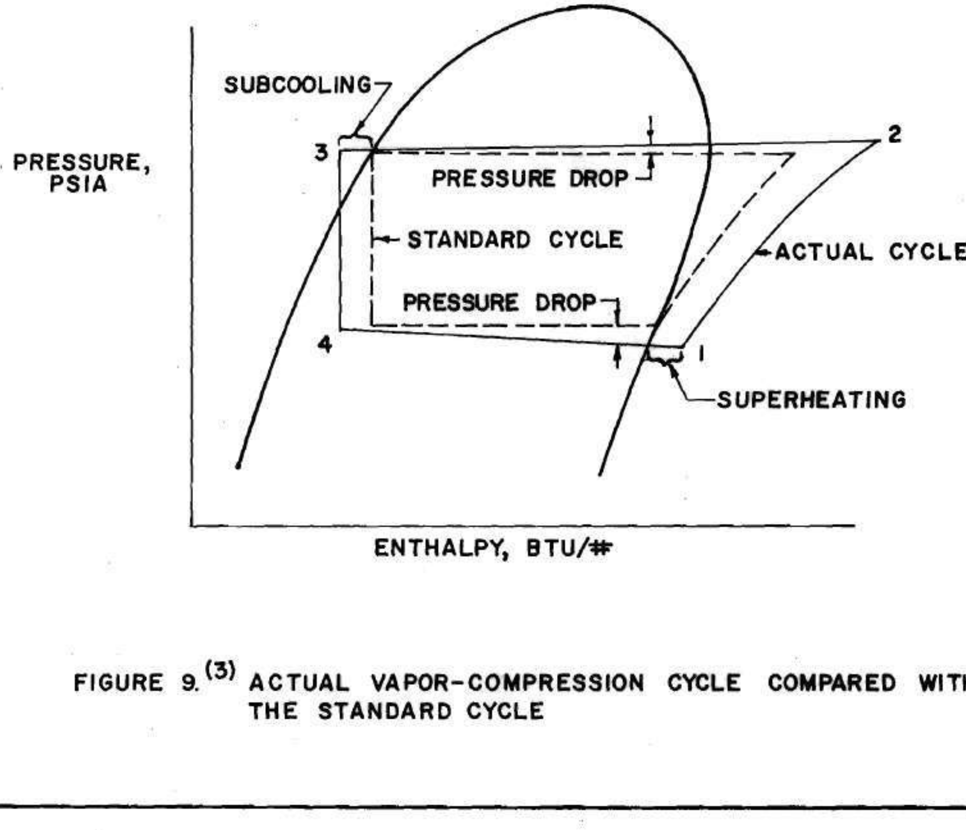


FIGURE 9.[3] ACTUAL VAPOR-COMPRESSION CYCLE COMPARED WITH THE STANDARD CYCLE

evaporator. Friction causes the pressure drops in the actual cycle, resulting in more required work in the compression process between points 1 and 2 than in the standard cycle. Subcooling of the liquid in the condenser is a normal occurrence and serves the desirable function of ensuring 100% liquid entering the expansion valve. Superheating of the vapor usually occurs in the evaporator and is recommended as a precaution against liquid droplets being carried into the compressor. Finally, the actual compression is not isentropic and inefficiencies due to friction and other losses occur [3].

Vapor-Compression Cycle Analysis

Consider, once again, the T-S diagram showing the thermodynamic path of the working fluid in the vapor-compression system (Figure 10). States 1, 2, 3, and 4 are the evaporator, compressor, condenser, and expansion valve exit conditions. State 1 is taken to be 5°F superheated and state 3 to be 5°F subcooled.

The principal disadvantage of superheating the compressor suction vapor is that the increased specific volume reduces the capacity of the compressor in terms of mass rate of circulation. Also, the top temperature of the cycle will be raised increasing the work of compression required. It is usually important, however, to avoid the admission of liquid phase fluid to the compressor, and

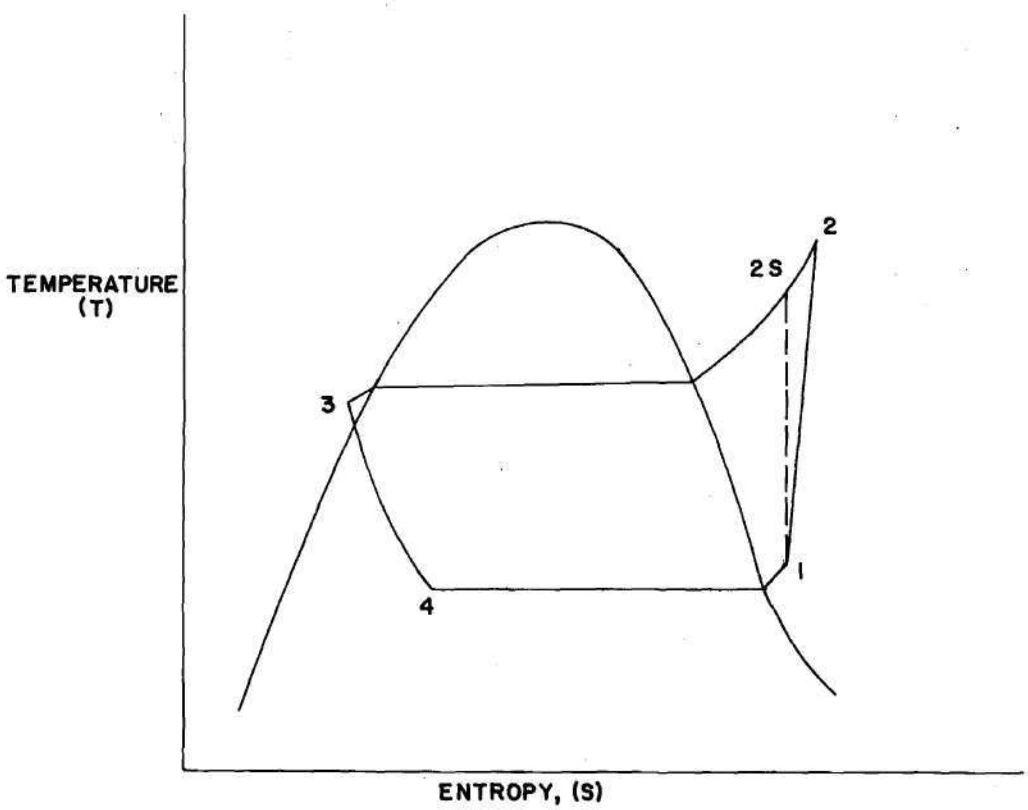


FIGURE 10.[1] WORKING FLUID T-S DIAGRAM

several degrees of superheat will ensure against this possibility.

Subcooling of the high-pressure liquid may increase the refrigerating effect. Also, if the actual system was designed to bring the fluid leaving the condenser only to the saturation line, any friction loss in the piping would result in throttling into the wet-vapor region. The resulting vapor formation and increase in specific volume would reduce the capacity of the piping, and, even more importantly, the capacity of the throttling device.

The ideal compressor work is given by:

$$\dot{W}_{ideal} = \dot{m}\ (h_{2s} - h_1) \tag{16}$$

where:

$\dot{m}$ = freon mass flow rate

h_{2s} = enthalpy at state 2S

h_1 = enthalpy at state 1

Inefficiencies reduce compressor performance, however, leading to a definition of compressor isentropic efficiency:

$$\eta_S = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{17}$$

The actual work is:

$$\dot{W}_{act} = \dot{m}\ (h_2 - h_1) \tag{18}$$

Substituting equation (17) into equation (18):

$$\dot{W}_{act} = \dot{m} \left(\frac{h_{2s} - h_1}{\eta_s}\right) \tag{19}$$

The heat rejected by the condenser is given by:

$$\dot{Q}_{cond} = \dot{m} \, (h_2 - h_3) \tag{20}$$

In the cooling mode, $\dot{Q}_{cond}$ is the amount of heat rejected to the atmosphere. In the heating mode, however, $\dot{Q}_{cond}$ is the desired heating effect.

The heat received by the evaporator is given by:

$$\dot{Q}_{evap} = \dot{m} \, (h_1 - h_4) \tag{21}$$

$\dot{Q}_{evap}$ represents the amount of heat taken from a home, for example, in the cooling mode, while it represents the heat gained from the atmosphere in the heating mode.

The mass flow rate is given by the equation:

$$\dot{m} = \rho \, \eta_v \, \dot{V} \tag{22}$$

where:

ρ = the vapor density at the compressor inlet

η_v = the compressor volumetric efficiency

$\dot{V}$ = the compressor piston displacement per unit time (given by the product of the compressor speed times piston displacement per revolution).

The heating mode has an additional feature for recovering heat. The exhaust gases from the engine can be drawn through a heat exchanger to provide the following waste heat recovery:

$$\dot{Q}_{rec} = F_{rec}\ \dot{Q}_{waste} = F_{rec}\ (1-\eta_{th})(\dot{W}_{comp}/\eta_{th}) \qquad (23)$$

where

F_{rec} = the fraction of waste heat recovered

η_{th} = the thermal efficiency of the engine

Therefore, the total heat available in the heating mode is:

$$\dot{Q}_{heating} = \dot{Q}_{cond} + \dot{Q}_{rec} \qquad (24)$$

The coefficient of performance (COP), the desired efficiency parameter for the gas heat pump, is the ratio of the desired heating or cooling output to the input energy of the gas. Therefore, this COP is based on the gas, or heat, input rather than the work input. For the cooling mode:

$$COPH_C = \frac{\dot{Q}_{evap}}{\dot{Q}_{gas}} \qquad (25)$$

Using:

$$\dot{Q}_{gas} = \frac{\dot{W}_{comp}}{\eta_{th}} \tag{26}$$

results in the following expression:

$$COPH_c = \frac{\dot{Q}_{evap}\ \eta_{th}}{\dot{W}_{comp}} = COPW_c\ \eta_{th} \tag{27}$$

For the heating mode:

$$COPH_H = \frac{\dot{Q}_{heating}}{\dot{Q}_{gas}} \tag{28}$$

Recalling:

$$\dot{Q}_{heating} = \dot{Q}_{cond} + \dot{Q}_{rec} \tag{29}$$

Substituting equation (23) into equation (29):

$$\dot{Q}_{heating} = \dot{Q}_{cond} + F_{rec}\ (1-\eta_{th})(\dot{W}_{comp}/\eta_{th}) \tag{30}$$

Substituting equation (30) and equation (26) into equation (28):

$$COPH_H = \frac{\dot{Q}_{cond} + F_{rec}\ (1-\eta_{th})(\dot{W}_{comp}/\eta_{th})}{\dot{W}_{comp}/\eta_{th}}$$

Simplifying:

$$COPH_H = \frac{\eta_{th}\ \dot{Q}_{cond}}{\dot{W}_{comp}} + F_{rec}\ (1-\eta_{th})$$

$$= \eta_{th}\ COPW_H + F_{rec}\ (1-\eta_{th}) \qquad (31)$$

Since the coefficient of performance of a system is a function of the temperatures under which the system operates (i.e. T_L and T_H), it would seem desirable to obtain a relationship between COP_c and COP_H and some parameter involving T_L and T_H. A logical choice are the Carnot or ideal COP's for a work-driven system. Recall:

$$COPW_c(\text{ideal}) = \left(\frac{T_L/T_H}{1-T_L/T_H}\right) \qquad (5)$$

and

$$COPW_H(\text{ideal}) = \left(\frac{1}{1-T_L/T_H}\right) \qquad (6)$$

In order to determine the desired relationships, consider the following four equations:

$$\text{Electric cooling:}\quad COPW_c = K_1\ \left(\frac{T_L/T_H}{1-T_L/T_H}\right) \qquad (32)$$

Gas cooling: $$COPH_c = K_1 \left(\frac{T_L/T_H}{1-T_L/T_H}\right) \tag{33}$$

Electric heating: $$COPW_H = K_1 \left(\frac{1}{1-T_L/T_H}\right) \tag{34}$$

Gas heating: $$COPH_H = K_1 \left(\frac{1}{1-T_L/T_H}\right) + K_2 \tag{35}$$

The values of K_1 and K_2 represent constants which can be determined. For the electrical system, K_1 is a function of losses such as the electric compressor inefficiency, the electric motor inefficiency, heat losses in piping, and other losses due to real fluid properties. Similarly for the gas system, K_1 is a function of system losses and component inefficiencies. It should be noted, however, that for the gas system, the maximum value that K_1 can attain is the gas engine efficiency. The constant K_2 is a function of the waste heat recovery, and does not appear in the electrical system analysis, of course, since in this case $F_{rec} = 0$.

A wide range of temperatures was chosen: for T_L = 20°F, 40°F, and 60°F, T_H was taken to be 80°F, 100°F, and 120°F. All nine combinations were considered. For these values, using Freon 22 properties, the COP's for ideal vapor-compression cycles were calculated as shown in Appendix A. In each case, an average of the nine calculated values for K_1 (and K_2) was taken as the constant in the relationship for an ideal system using a real refrigerant. Figures 11, 12, 13, and 14 present the results. Note that $COPW_H$(ideal)

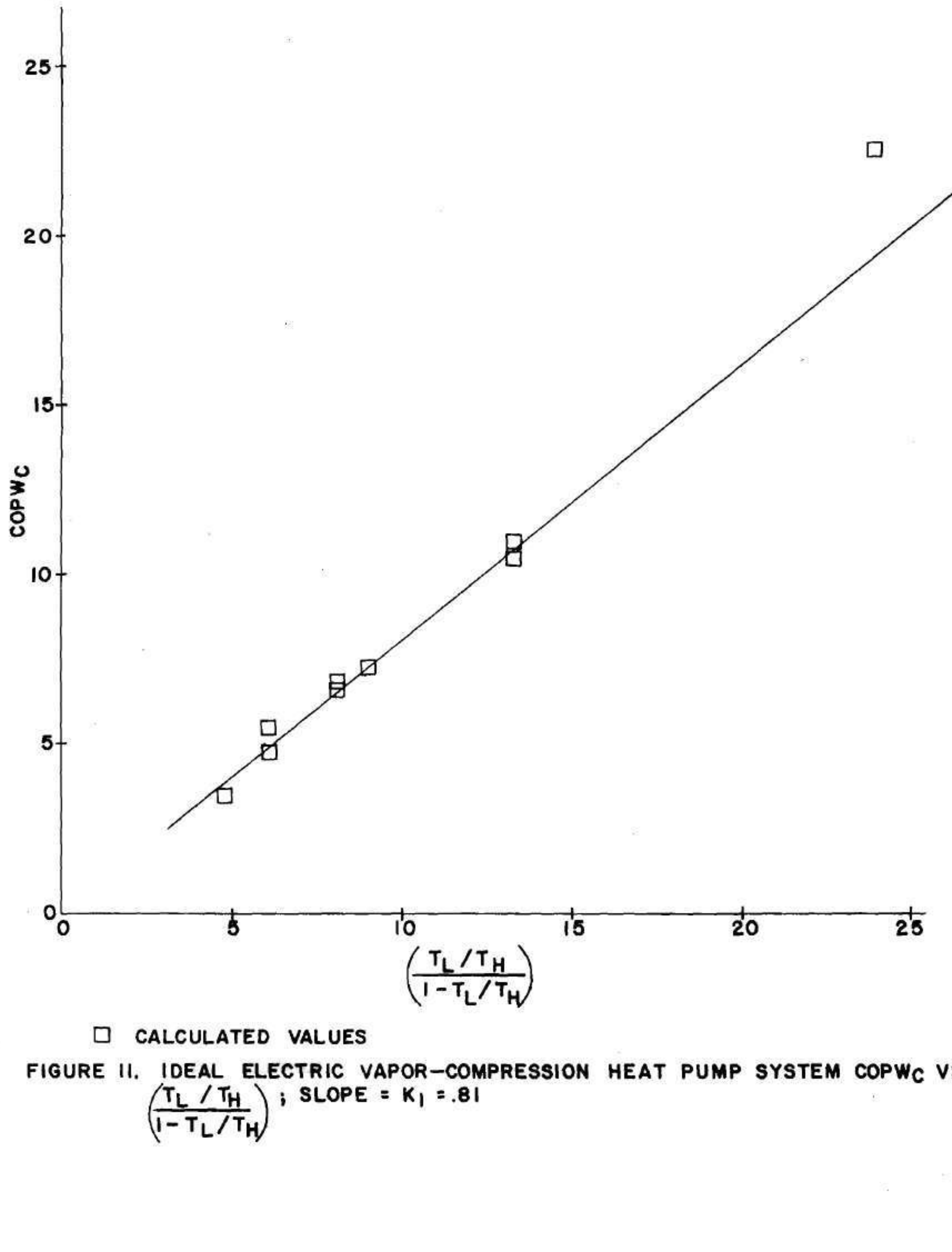


FIGURE 11. IDEAL ELECTRIC VAPOR–COMPRESSION HEAT PUMP SYSTEM $COPW_C$ VS. $\left(\frac{T_L/T_H}{1-T_L/T_H}\right)$; SLOPE = K_1 = .81

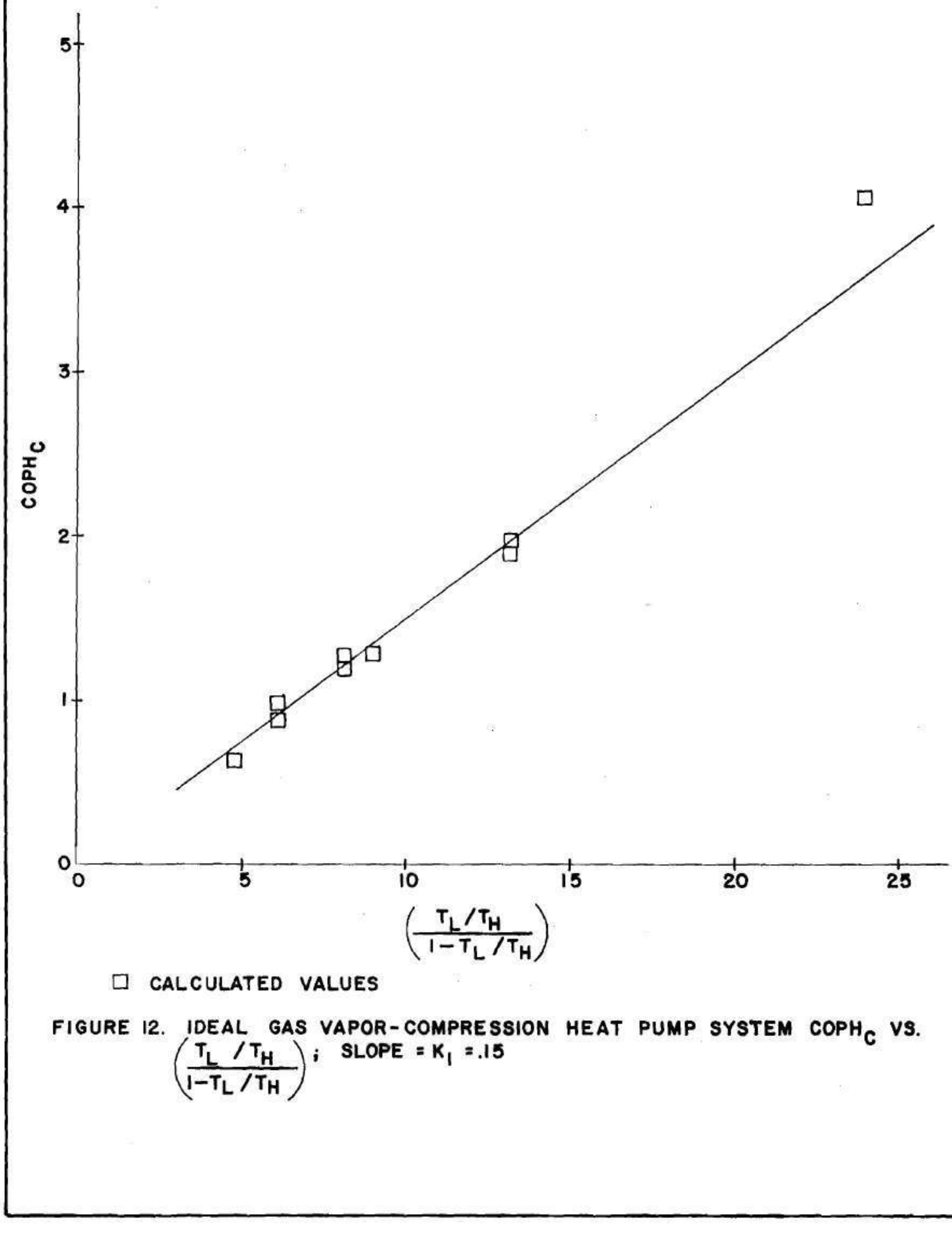


FIGURE 12. IDEAL GAS VAPOR-COMPRESSION HEAT PUMP SYSTEM COPH$_C$ VS. $\left(\frac{T_L/T_H}{1-T_L/T_H}\right)$; SLOPE = K_1 = .15

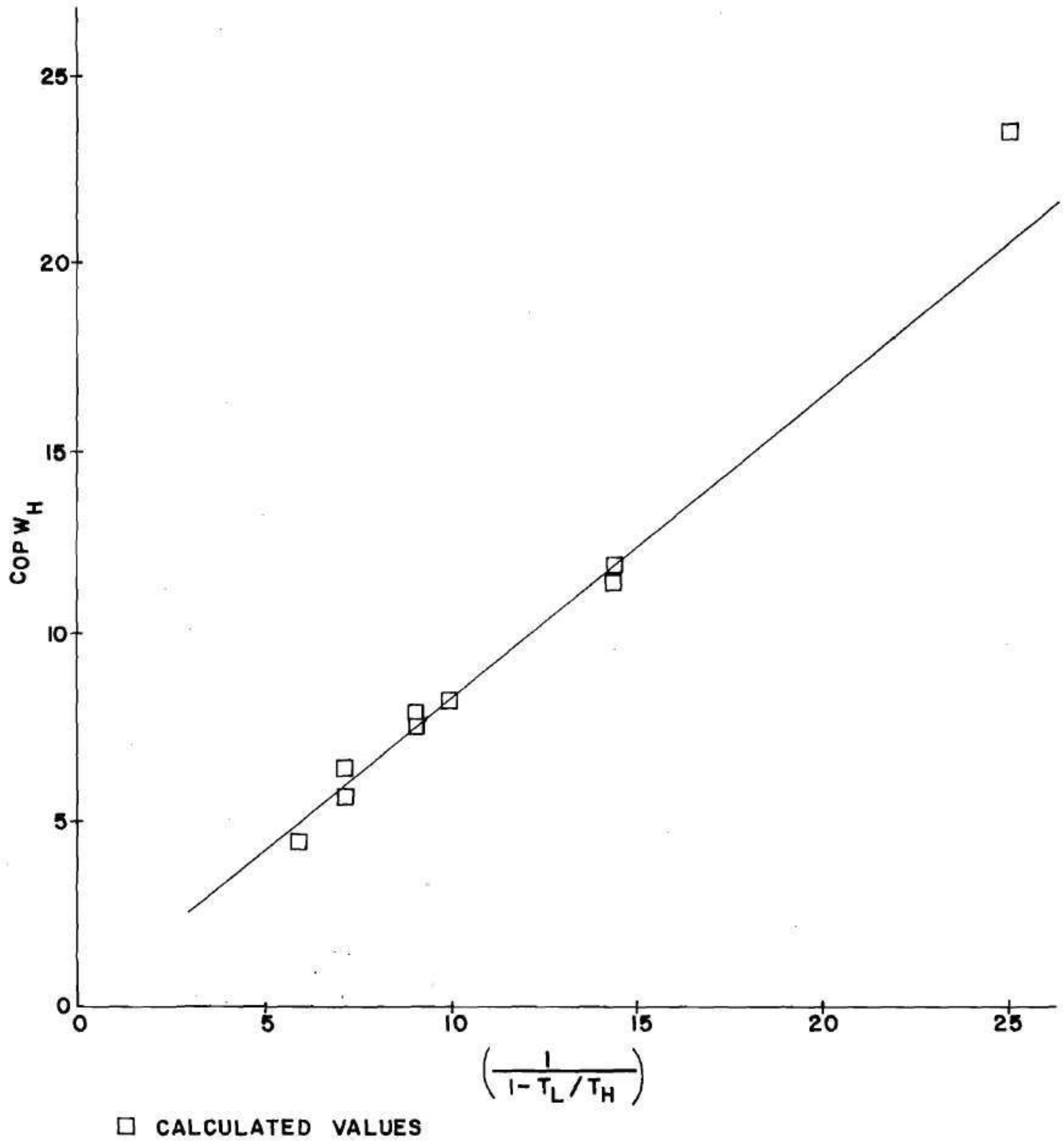


FIGURE 13. IDEAL ELECTRIC VAPOR-COMPRESSION HEAT PUMP SYSTEM COPW_H VS. $\left(\frac{1}{1-T_L/T_H}\right)$; SLOPE = K_1 = .83

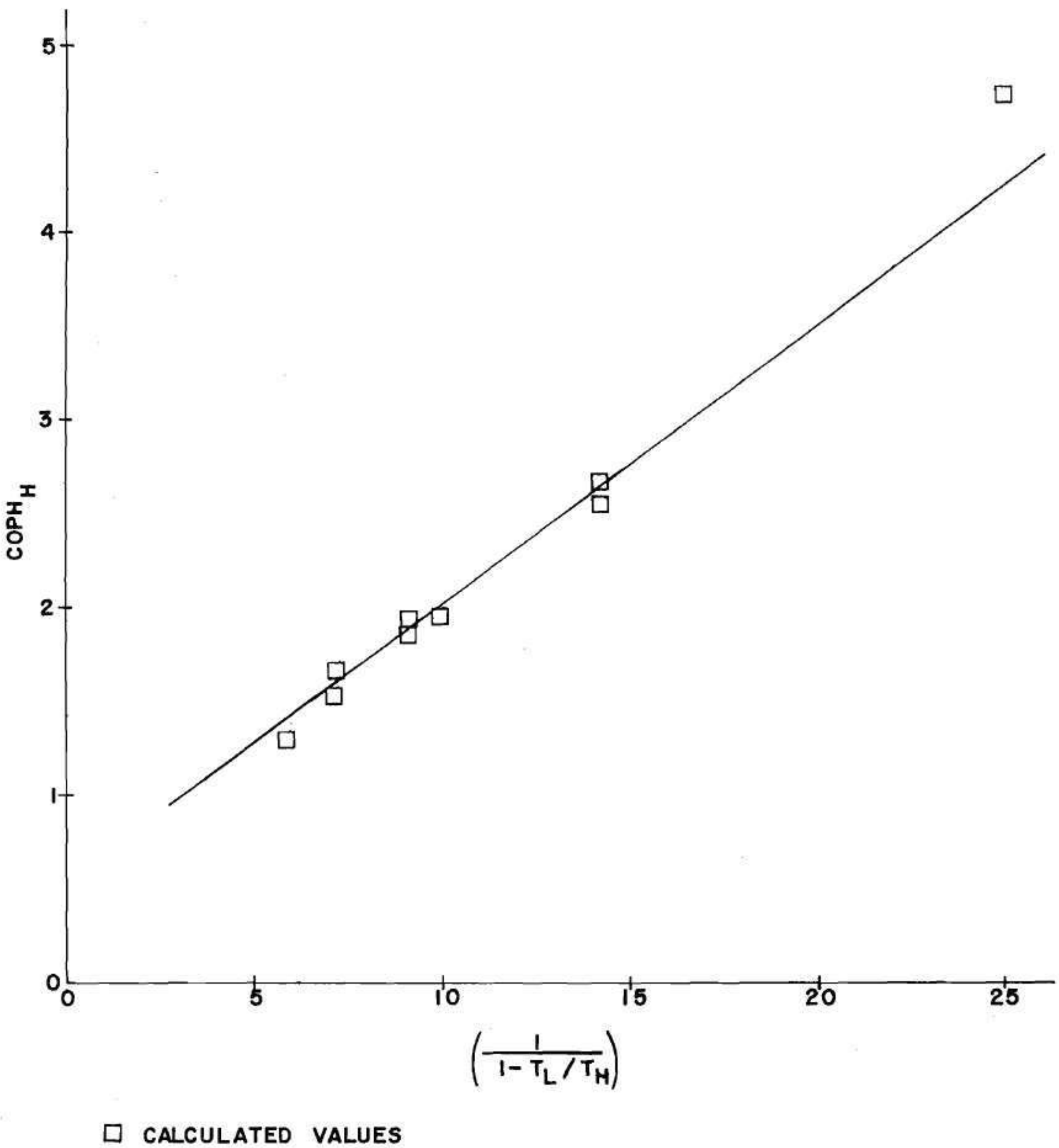


FIGURE 14. IDEAL GAS VAPOR-COMPRESSION HEAT PUMP SYSTEM WITH WASTE HEAT RECOVERY $COPH_H$ VS. $\left(\frac{1}{1-T_L/T_H}\right)$; SLOPE = K_1 = .15; K_2 = .49

approaches 1.0 as a lower limit, while $COPW_c$(ideal) approaches 0.0 as a lower limit. For the gas system, an engine efficiency of 18% with F_{rec} = .60 was assumed. Thus, in Figure 14, at $(\frac{1}{1-T_L/T_H}) = 1.0$, $COPH_H$ should equal .18 + K_2 or 0.67.

Once again, Figures 11, 12, 13, and 14 present the relationships between the ideal COPW's and the COP's for systems operating on an ideal vapor-compression cycle but using a real working fluid, Freon 22. Later in this work, similar relationships will be determined between the ideal COPW's and the COP's for the actual systems studied.

Annual Average COPH

The coefficient of performance varies with temperature which, of course, varies throughout the year. It would be desirable, therefore, to determine an annual average COP. The following allows calculation of such a value, given a certain temperature, and thus, a particular COPH.

The annual average $COPH_c$ and $COPH_H$ are given by:

$$\overline{COPH}_c = \overline{Q}_c/\overline{Q_{gas}} \tag{36}$$

$$\overline{COPH}_H = \overline{Q}_H/\overline{Q_{gas}} \tag{37}$$

where:

$(\overline{})$ denotes annual quantities

Q_{gas} = the heating value of the natural gas used

The heating and cooling required by a residence is proportional to the temperature difference between 65°F and the outside air temperature. Therefore:

$$\dot{Q}_C = K(T_A - 65);\ T_A > 65°F \tag{38}$$

$$\dot{Q}_H = K(65 - T_A);\ T_A < 65°F \tag{39}$$

where:

T_A = the outside air temperature in °F

K = a constant, Btu/hr-°F (a typical value for K for the city of Atlanta is 1,000 Btu/hr-°F for a 1500 ft^2 home).

When T_A = 65°F, enough heat is supplied by indoor appliances to keep the indoor temperature within the comfort region. Therefore:

$$\overline{Q}_C = \int_0^{t_c} K(T-65)dt;\ T > 65 \tag{40}$$

$$\overline{Q}_H = \int_0^{t_h} K(65-T)dt;\ T < 65 \tag{41}$$

where:

t = time

t_c = the annual cooling period

t_H = the annual heating period

The annual gas demand for cooling is:

$$\overline{Q_{gas}} = \int_{o}^{t_c} \frac{K(T-65)}{COPH_c}\, dt;\ T > 65 \tag{42}$$

Similarly for heating:

$$\overline{Q_{gas}} = \int_{o}^{t_H} \frac{K(65-T)}{COPH_H}\, dt;\ T < 65 \tag{43}$$

Note that $COPH_c$ and $COPH_H$ in equations (42) and (43) are not constant, but vary with temperature. Making the appropriate substitutions results in the following:

$$\overline{COPH_c} = \frac{\int_{o}^{t_c} (T-65°F)dt}{\int_{o}^{t_c} \frac{(T-65°F)}{COPH_c}\, dt} \tag{44}$$

$$\overline{COPH_H} = \frac{\int_{o}^{t_h} (65°F-T)dt}{\int_{o}^{t_h} \frac{(65°F-T)}{COPH_H}\, dt} \tag{45}$$

These integrals can be evaluated using hourly temperature data for a given year. This data for Atlanta, Georgia, for the year 1964, is presented in reference 1 and reproduced below. The number of hours annually at 10°F temperature intervals were found for this year to be [1]:

°F	Hours/Year
0 - 9	0
10 - 19	25
20 - 29	240
30 - 39	851
40 - 49	1417
50 - 59	1433
60 - 69	1792
70 - 79	2177
80 - 89	848
90 - 99	77

Information of this type, along with $COPH_c$ and $COPH_H$, expressed as a function of outside temperature, allows evaluation of the $\overline{COPH}_c$ and $\overline{COPH}_H$ in equations (44) and (45).

CHAPTER III

GAS FURNACE STUDY

Equipment and Principle of Operation

Initially a study was made of a gas furnace type of heater manufactured by Amana Refrigeration, Inc. The heating cycle studied is accomplished by an indirect method in which a gas flame heats a circulating liquid in a Heat Transfer Module. The heated liquid (50% Ethylene Glycol and 50% distilled water) is circulated by a pump through an indoor coil. Air from the conditioned space is drawn across the indoor coil by the blower motor where it is warmed and returned to the conditioned space (see Figure 15).

The liquid heating is accomplished by using a unique, compact heat exchanger. This unit (Heat Transfer Module) consists of multiple tubes (24 passage tubes arranged so that six passes are made through the module) embedded in a nickel-plated steel ball matrix, which is oven brazed. Through this ball construction pass the products of combustion. The gas burner, located inside the steel ball matrix, consists of a stainless steel cylindrical screen with approximately 9,980 holes. The gas-air mixture burns on the outside of the screen with very minute flames.

A motor operated combustion blower is used to draw

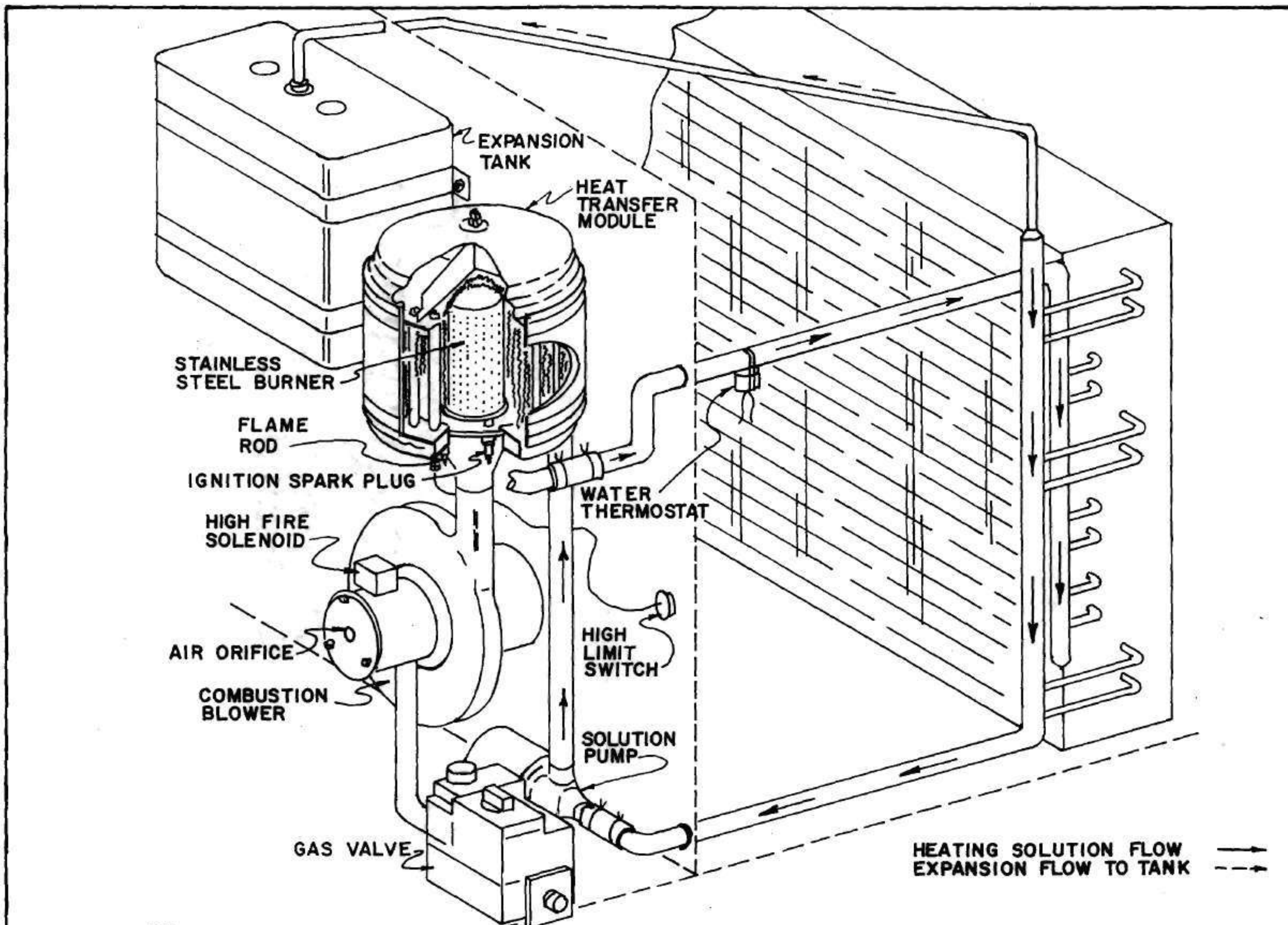


FIGURE 15.[4] SCHEMATIC HEATING SYSTEM

the gas supply from the electrically operated gas valve, at a negative pressure (below atmospheric), through a predetermined gas orifice into the mixing chamber. Primary combustion air is also drawn through a matching predetermined air orifice, to be mixed with the gas supply by the combustion blower wheel. This gas-air mixture is then discharged into the stainless steel gas burner. The mixture is ignited by the use of a specially designed spark ignitor, which is powered electrically by a direct spark ignition control. The products of combustion are power vented to the atmosphere through the flue outlet and vent cap at the side of the unit.

The heating solution, consisting of a 50% mixture of Ethylene Glycol (with corrosion inhibitor) and distilled water, is circulated by a small motor driven centrifugal pump, at a velocity of approximately 4.77 feet/second, giving turbulent flow in the heat exchanger tubes for a high rate of heat transfer. The solution circulates through the heat exchanger where it is heated and then delivered to the indoor heating coil (see Table 4). Indoor circulated air is passed over the coil by the indoor blower motor, to remove the heat and discharge it into the conditioned space. The cooled heating solution is then returned back to the water pump inlet, ready to repeat the cycle. The heating solution is circulated at approximately eight gallons/minute.

The system operates at essentially atmospheric pressure. Since the solution expands when heated and contracts when

Table 4. Indoor Heating Coil Specifications for the Gas Furnace System [5]

Face Area:	2.71 ft^2
Rows Deep:	4
Fins/in:	12
Tubes, O.D. inches:	3/8

cooled, an expansion tank is provided that is open to the atmosphere using a split rubber grommet to control the evaporation rate. This also provides a point where any air that is in the system escapes during the heating-up period. Between cycles, as the solution cools, it is siphoned out of the expansion tank back into the system.

Instrumentation

Among the parameters of importance were the water temperature before and after the heat exchanger, the water flow rate, the natural gas input, and the electrical input (see Figure 16). An iron-constantan thermocouple was attached to the surface of the water line going to the heat exchanger. Similarly, a thermocouple was attached to the return water line. The lines were then carefully insulated to reduce heat loss. The temperatures were measured on an Omega Engineering, Inc. pyrometer, designed to read directly in degrees Fahrenheit with an accuracy of ±2%. The water flow rate was measured on a standard Neptune water meter installed in the return line. The gas input was determined on a

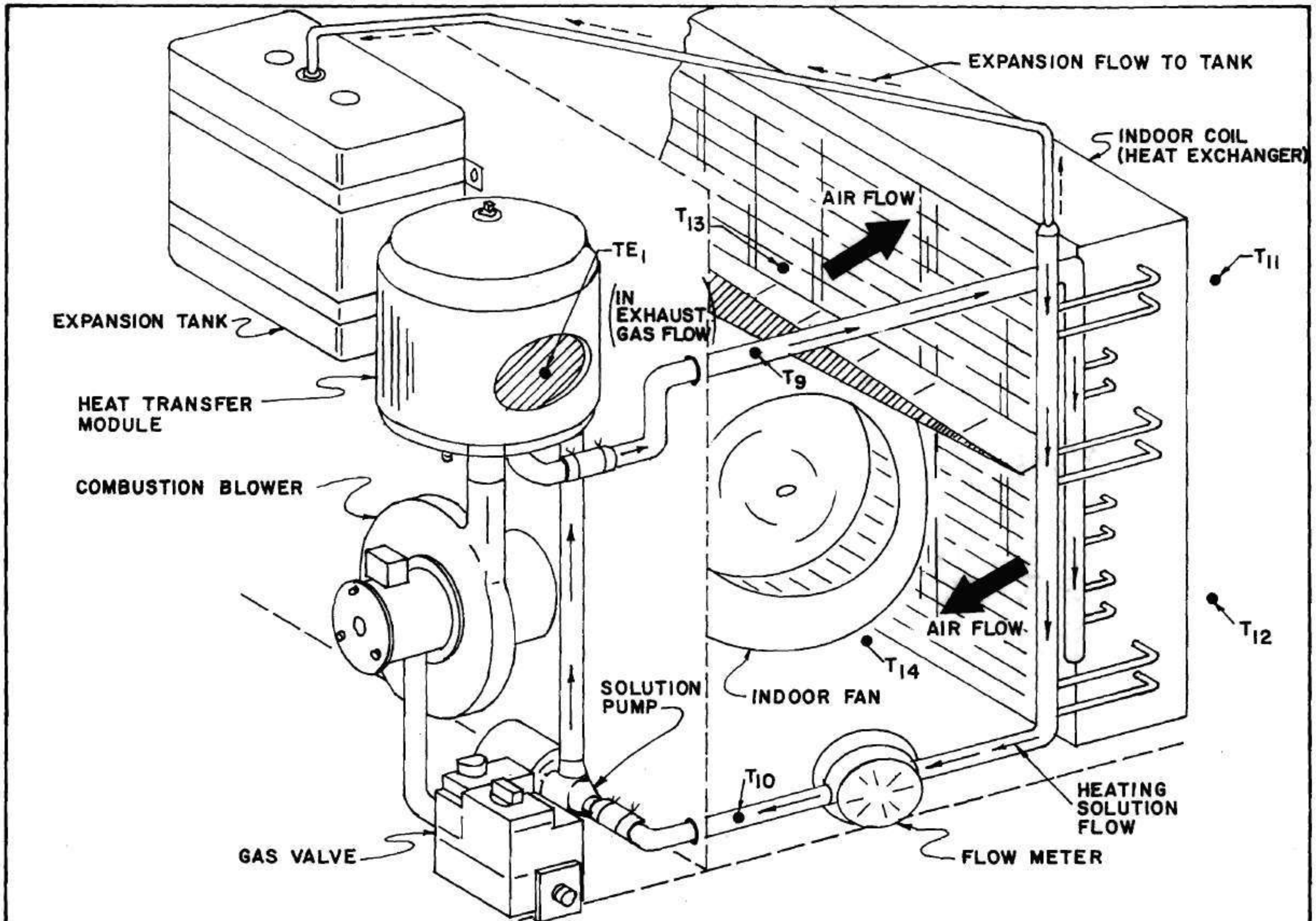


FIGURE 16. SCHEMATIC OF THE GAS FURNACE HEATING SYSTEM INDICATING THE LOCATION OF THE FLOW METER AND THERMOCOUPLES

typical residential gas meter, while a General Electric single-phase watthour meter, type 1-50-S, allowed measurement of the electrical input.

Test Results

The primary value of interest was the efficiency, defined as follows:

$$\eta = \frac{\dot{m}_w \ \Delta T_w \ C_{pw}}{\dot{Q}_{gas}} \tag{46}$$

where:

$\dot{m}_w$ = the mass flow rate of water, #m/min

ΔT_w = the water temperature difference across the heat exchanger, °F

C_{pw} = the water specific heat, 1 Btu/(#m°F)

$\dot{Q}_{gas}$ = energy content of the gas being input, $\dot{Q}_{gas}$ = ($\dot{V}$, the volume flow rate, ft^3/min) X (the heating value of methane = 1,030 Btu/ft^3)

Four combinations were tested: a high and a low heat setting, referring to the rate of natural gas and air intake and subsequent combustion, and a high and a low fan, referring to the indoor fan speed setting. Appendix B contains a table of the data collected, and a sample η calculation. The results are presented below:

low heat low fan	low heat high fan	high heat low fan	high heat high fan
η = 80.3%	η = 84.0%	η = 86.4%	η = 84.8%

From their literature search, Calvert and Harden [6] found that the $COPH_H$ for a gas furnace can be taken to be 0.7 [1]. The term which is defined as η above is equivalent to $COPH_H$. Needless to say, the particular machine analyzed here performed better than a conventional gas furnace. This improvement is due to the efficient heat exchanger on this unit.

A gas furnace provides a very effective means of home heating. It is limited, however, to just that--heating. If summer cooling is required in addition to winter heating an additional cooling system must be provided. Herein lies one advantage of a heat pump system in which the same components can be used for both heating and cooling. Moreover, as shown in the introduction, a gas heat pump in Atlanta, Georgia, would heat and cool year around with less total gas than a gas furnace would require for heating only. The conclusion should be obvious. Although the test results for this system are included in the electric heat pump--gas heat pump comparison presented later in this work, a gas furnace alone is not the answer to total home environmental control.

CHAPTER IV

ELECTRIC HEAT PUMP STUDY

Equipment and Principle of Operation

A heat pump, electric or gas, is a system which pumps heat from a low temperature to a higher temperature, with a necessary input of energy. This energy is usually supplied in the form of rotating shaft work to a compressor. If the work input to the system is provided by an electric motor, an electric heat pump results. An Amana five-ton (cooling) Central System Air Conditioner was the unit studied in this work. System schematics are presented in Figures 17, 18, and 19.

A refrigerating or heat pump cycle is made possible by a few basic principles [7]:

(1) Heat flows from a warmer object to a cooler one. Thus something can be cooled by placing it next to something even cooler.

(2) The temperature at which a liquid boils depends upon the pressure exerted on the liquid. The boiling point of the liquid can be controlled if the pressure on it is controlled. Thus, the boiling point of a liquid can be lowered to a point below the temperature of the object to be cooled by the liquid.

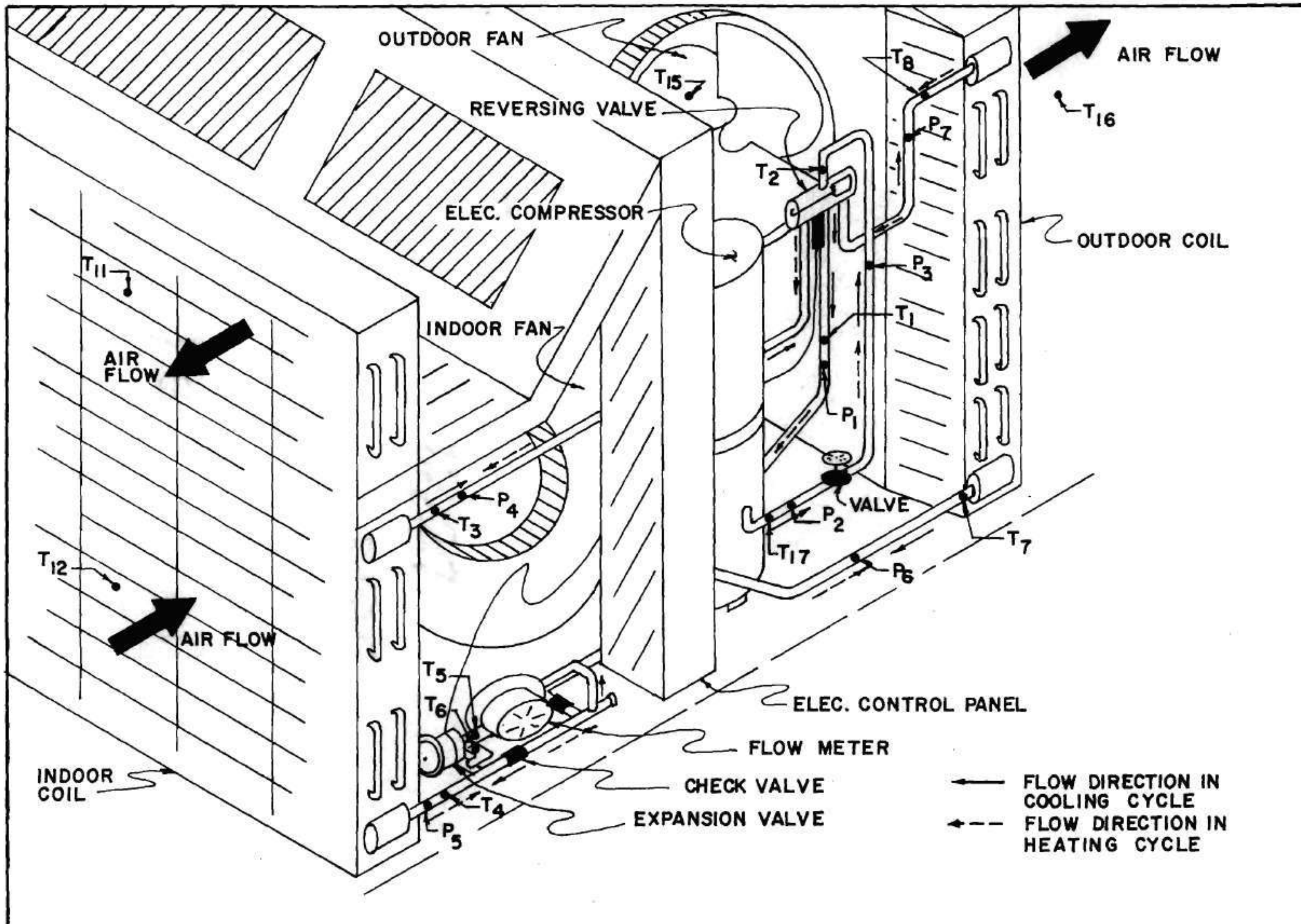


FIGURE 17. ELECTRIC HEAT PUMP SYSTEM SCHEMATIC INDICATING THE LOCATION OF MEASUREMENT POINTS

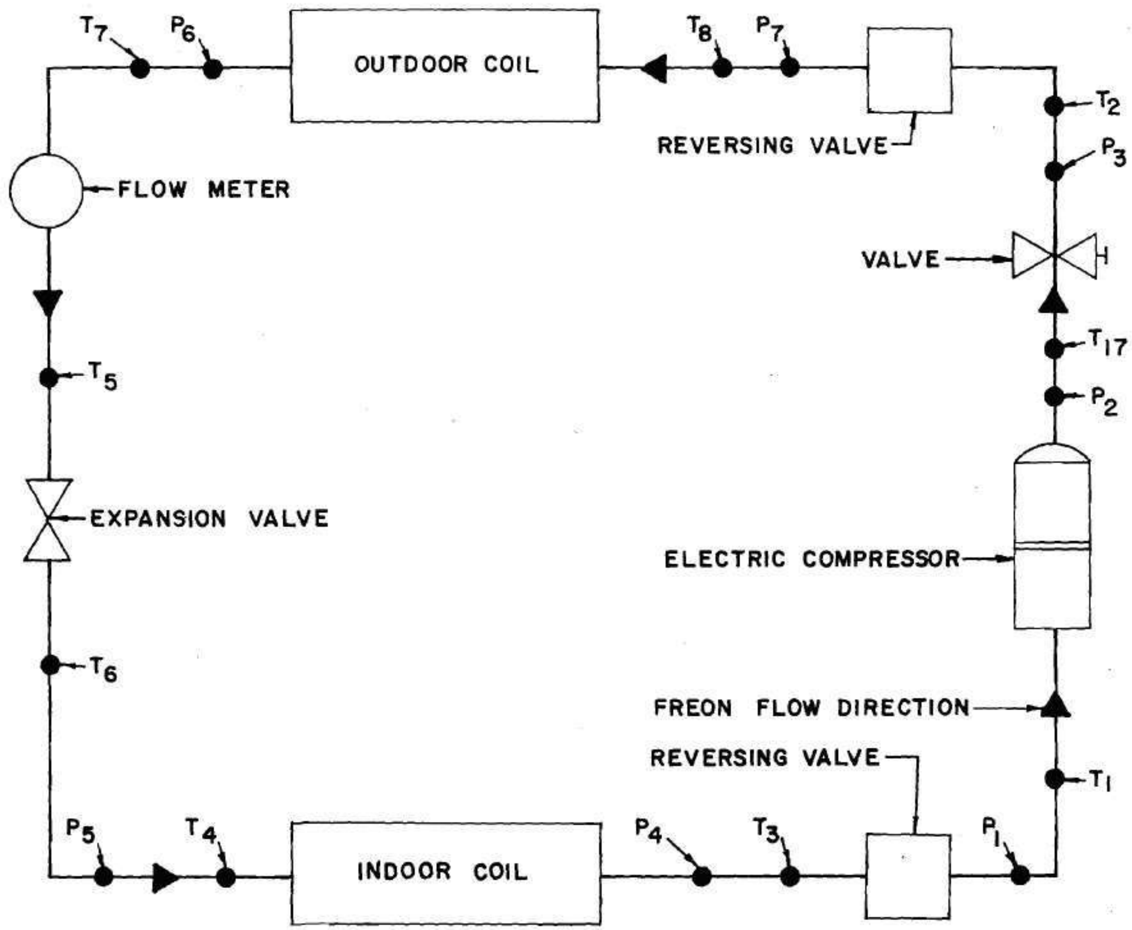


FIGURE 18. SCHEMATIC OF THE ELECTRIC HEAT PUMP COOLING CYCLE INDICATING THE LOCATION OF THE SIGNIFICANT MEASUREMENT POINTS

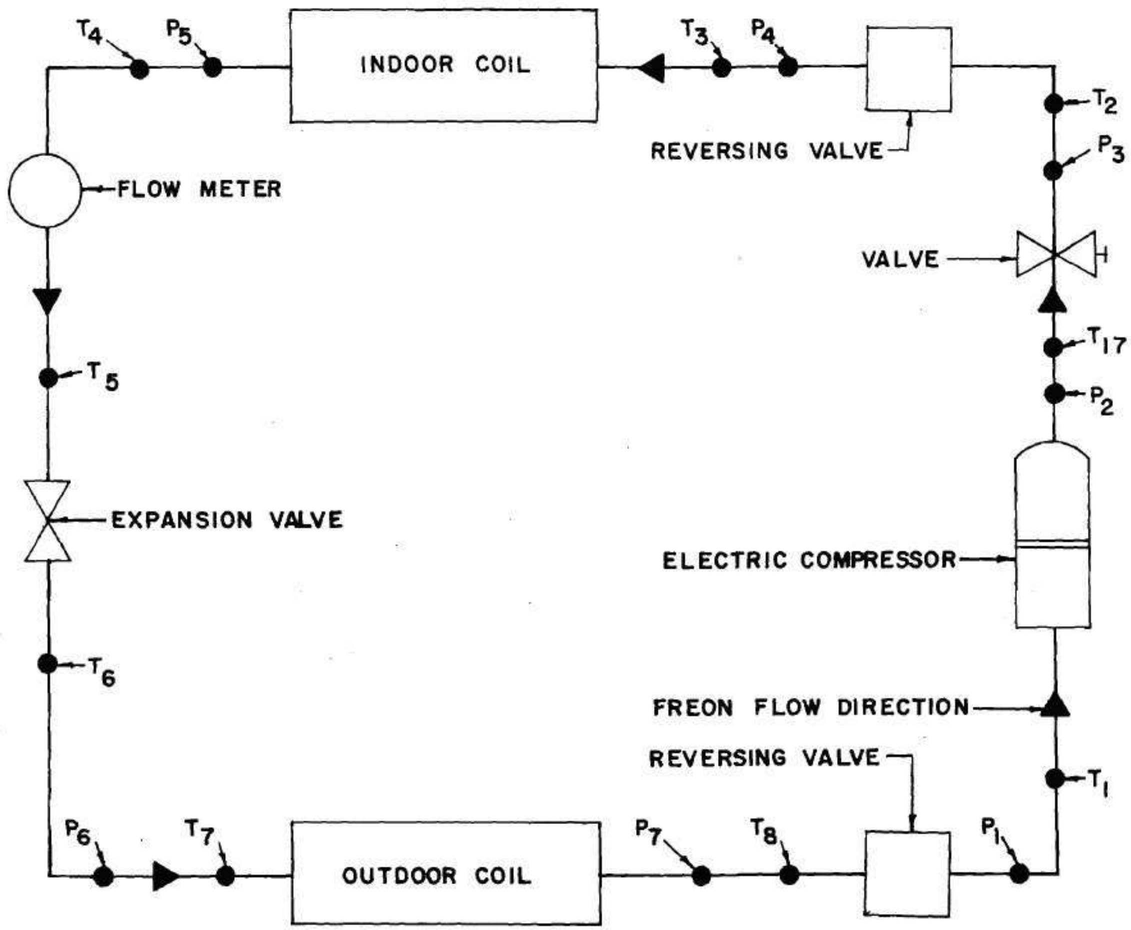


FIGURE 19. SCHEMATIC OF THE ELECTRIC HEAT PUMP HEATING CYCLE INDICATING THE LOCATION OF THE SIGNIFICANT MEASUREMENT POINTS

(3) A boiling liquid continually absorbs heat from any object placed near it. The temperature of the liquid does not rise, as the heat that is being absorbed is expended in vaporizing the liquid (latent heat of vaporization). Thus, continuous refrigeration is possible as long as there is liquid available for vaporization.

(4) Removing heat from a vapor causes the vapor to condense. Therefore, vapor can be returned to its liquid form after the cooling function has been performed by removing the heat that caused the vaporization. The vapor condensation temperature depends upon the pressure exerted on the vapor. If the pressure exerted on the vapor is high enough, the temperature of the vapor can be raised to a level where the vapor is hotter than the surrounding air. Heat will then flow from the hot vapor to the cooler surrounding air, and condensation of the vapor to a liquid will occur.

The vapor-compression cycle is one in which the working fluid alternately evaporates and condenses, with one of the intervening processes being a compression of the vapor. Refer to Figure 20.

The heart of the vapor-compression system is the compressor. Its function is two-fold. It must withdraw the fluid from the evaporator at a rate sufficient to maintain the necessary reduced pressure and temperature in the evaporator. In addition, it must compress and deliver the fluid at a temperature which is adequately above the substance

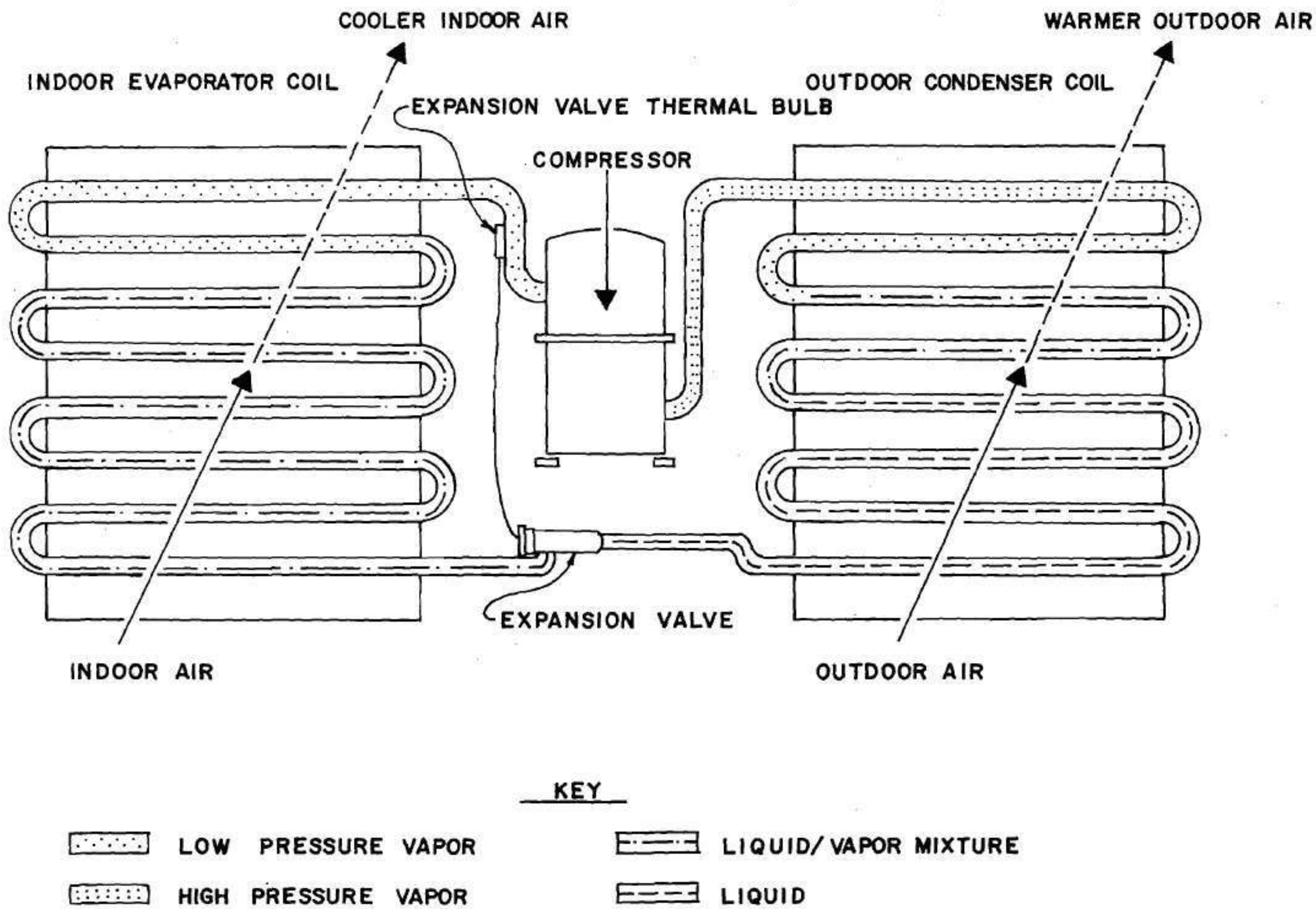


FIGURE 20. SCHEMATIC REFRIGERATION SYSTEM

to which the fluid must transfer its energy. The increase of temperature of the vapor must be accompanied by an increase of pressure.

The unit used in the electrical heat pump study was a five horsepower Tecumseh hermetic compressor (a direct-connected motor reciprocating-compressor assembly enclosed within a steel housing). Figure 21 presents some performance information supplied by the manufacturer [8].

After leaving the compressor the vapor enters the condenser in which it must first be desuperheated and then condensed, energy departing in the form of heat. The temperature during the condensation must moderately exceed that of the cooling air in order that the heat transition may proceed in the desired direction. The refrigerant pressure which must be delivered by the compressor is the saturation pressure corresponding to the condensing temperature. See Table 5 for the outdoor coil specifications (condenser in the cooling cycle).

Table 5. Outdoor Coil Specifications for the Heat Pump Systems [5]

Face Area:	7.49 ft^2
Rows Deep:	4
Fins/in:	12
Tube, O.D. inches:	3/8

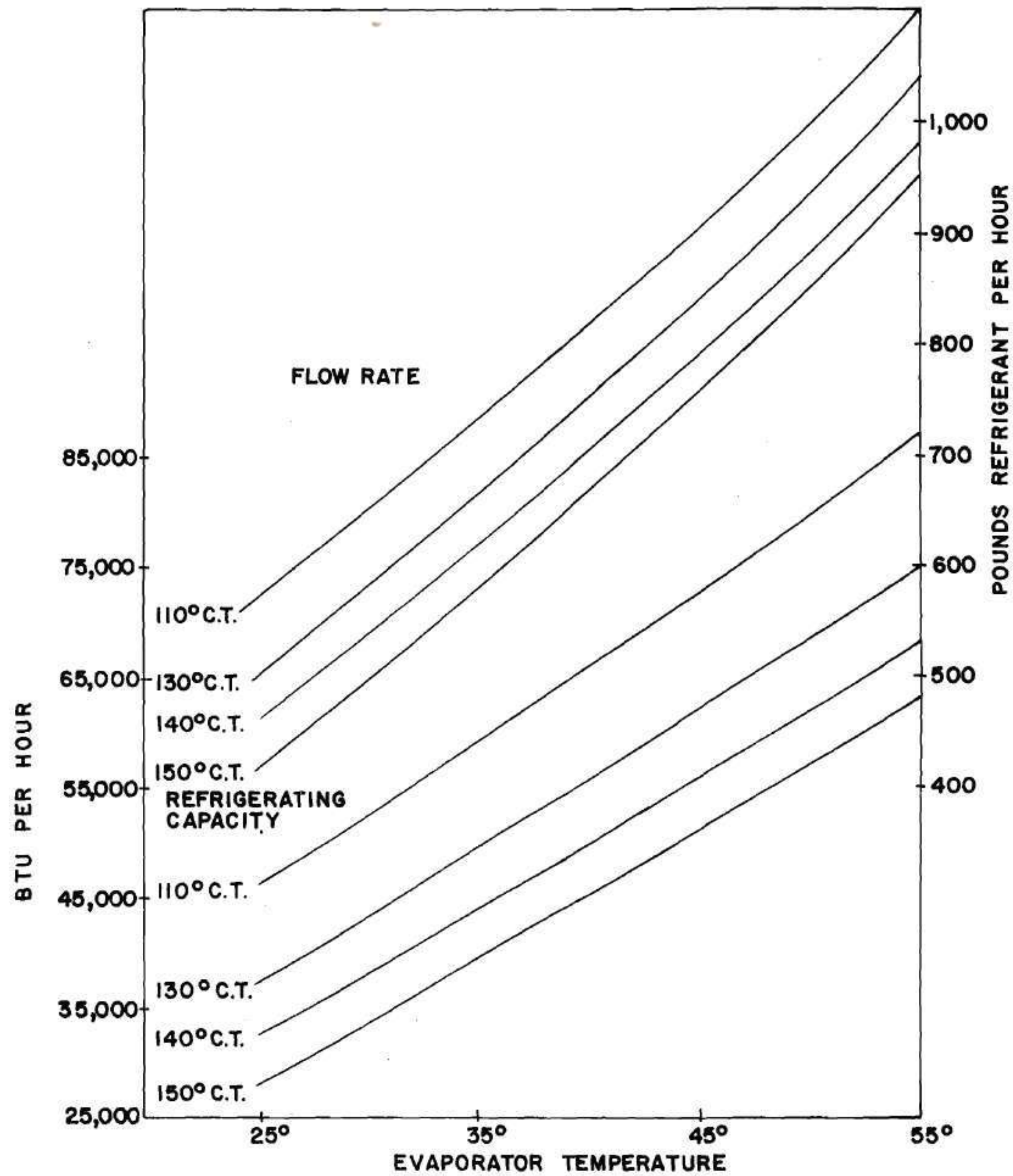


FIGURE 21B.[8] TECUMSEH COMPRESSOR PERFORMANCE CURVES (SEE TABLE 6 FOR SPECIFICATIONS)

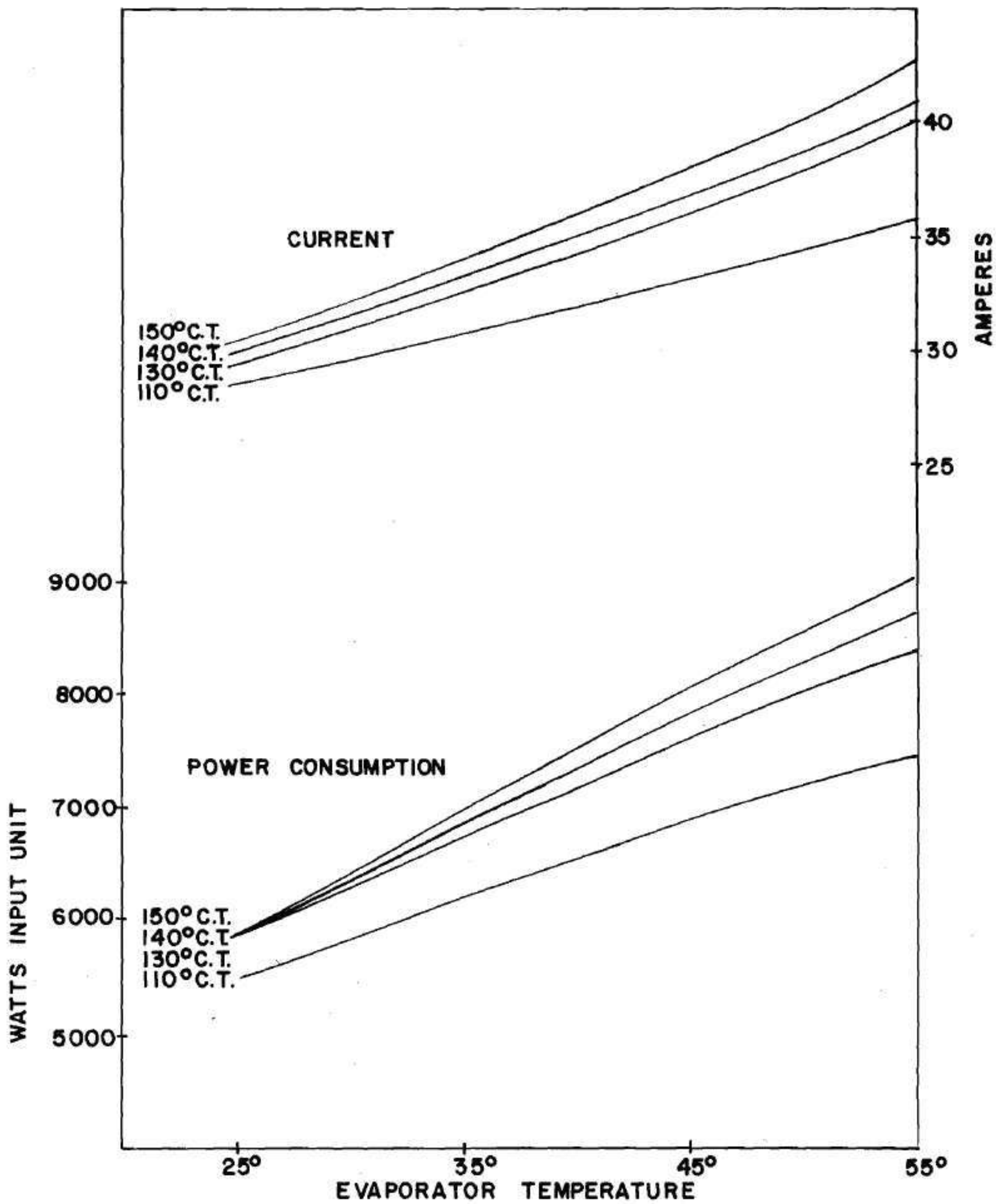


FIGURE 21A.[8] TECUMSEH COMPRESSOR PERFORMANCE CURVES (SEE TABLE 6 FOR SPECIFICATIONS)

Table 6. Specifications for the Tecumseh Compressor of Figure 21 [8]

Model:	CL5562E
Displacement:	7.24 in^3/ crankshaft revolution
Motor Type:	P.S.C.
Volts/Hz/Ph:	230/208-60-1
Tested at:	230 V.
Run Capacitor:	55 MFD
Room Ambient:	95°F
All Temperatures:	°F
Refrigerant:	R-22
Return Gas and Gas Leaving Evaporator Superheated to:	95°F
Forced Air Over the Compressor	
Liquid Subcooled 15° for all Condensing Temperatures: (C.T.)	

The working fluid next reaches the expansion valve, the purpose of which is also twofold. It must reduce the pressure of the refrigerant, and also must regulate the refrigerant flow to the evaporator. A turbine is not used because appreciable work cannot usually be recovered by the expansion of a very wet vapor through such a machine. Due to the throttling character of this process the fluid enthalpy before and after the device must be equal. A sensible portion of the liquid must vaporize and owing to the lower pressure of the vapor, its temperature must have fallen. Specifically, it must drop to the saturation temperature corresponding to the lower pressure.

A thermostatic expansion valve was installed in the unit replacing the existing capillary tube. The name "thermostatic" may be misleading because control is actuated not by the temperature in the evaporator, but by the amount of superheat of the suction gas leaving the evaporator. Designed to regulate the rate of liquid refrigerant flow into an evaporator in exact proportion to the rate of evaporation of the liquid refrigerant, the thermostatic expansion valve responds to both the temperature of the refrigerant gas leaving the evaporator, and the pressure in the evaporator.

After leaving the expansion valve the low temperature and low quality vapor mixture enters the evaporator. Here absorption of energy as heat by the working fluid from the region to be refrigerated is accomplished. Evaporation of the refrigerant results. The degree of completeness of its evaporation depends on the rapidity of its circulation, on the temperature difference between the fluid and its environment, and on other operating conditions. The fluid temperature in the evaporator must be maintained below the temperature of the cold region to ensure heat transfer in the desired direction. The pressure maintained must be the saturation pressure corresponding to the temperature. From the evaporator, the working fluid passes into the compressor to repeat the cycle. See Table 7 for the indoor coil specifications (evaporator in the cooling cycle).

Table 7. Indoor Coil Specifications for the Heat Pump Systems [5]

Face Area:	4.75 ft^2
Rows Deep:	4
Fins/in:	13
Tube, O.D. inches:	3/8

The heating cycle uses the same components as the cooling cycle just described and the principle of operation is the same with one exception. The functions of the evaporator and condenser are interchanged. Instead of physically exchanging the two heat exchangers, it is easier to simply reverse the direction of flow of the working fluid. Since the compressor is a one-way device, the most feasible method for changing flow direction is a reversing valve. A four-way, two-position reversing valve, operated by a three-way pilot solenoid valve, was installed in the system. See Figures 22 and 23.

The purpose of the pilot valve is to initiate the operation of the "sliding port" assembly in the main valve body, thus determining the refrigerant route through the evaporator and condenser coils, and thereby exchanging the functions of these two components to provide heating or cooling operation. A "sliding port" assembly, actuated by means of a piston arrangement, is operated by the pressure differential between the high and low sides of the refrigeration sytem. When the solenoid is energized, the right port in the pilot valve is opened, permitting the gas in the right-hand chamber of the main valve to bleed off through the capillary tube, thereby creating a pressure differential between the right and left-hand chambers. The higher pressure in the left-hand chamber causes the piston to move the "sliding port" assembly to the right producing the proper routing for the heating phase. When the solenoid is de-energized, the process is reversed. The left-hand port of the pilot valve opens, and the right port closes, allowing the pressure in the left-hand chamber to reduce. This causes the higher pressure of the right-hand chamber to move the "sliding port" to the left where it covers the opposite pair of tubes, resulting in the cooling phase.

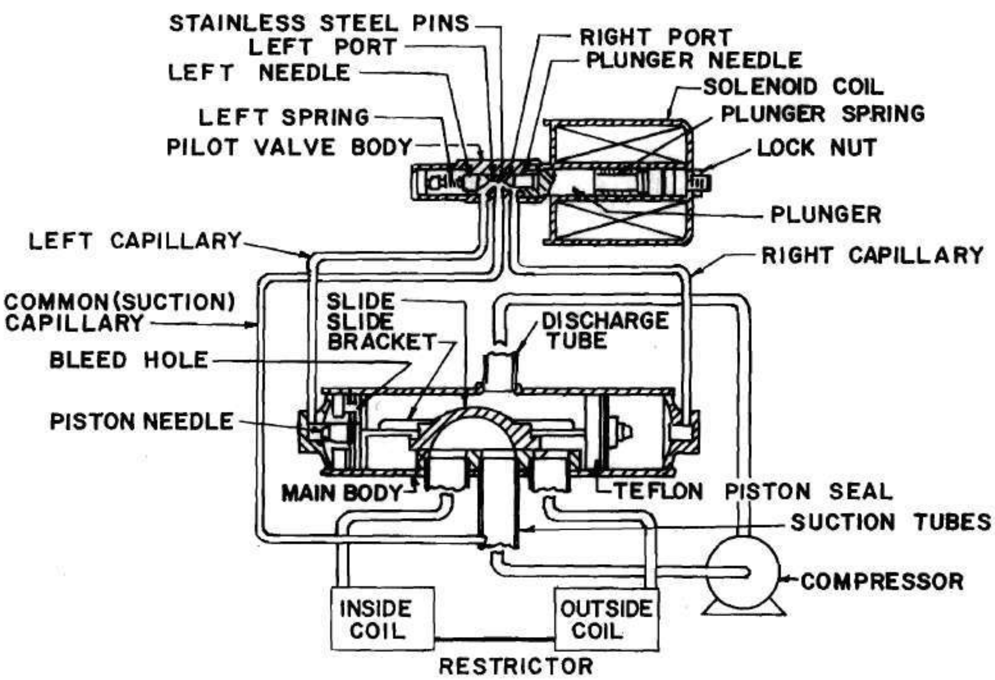


FIGURE 22.[9] SECTION VIEW OF REVERSING VALVE

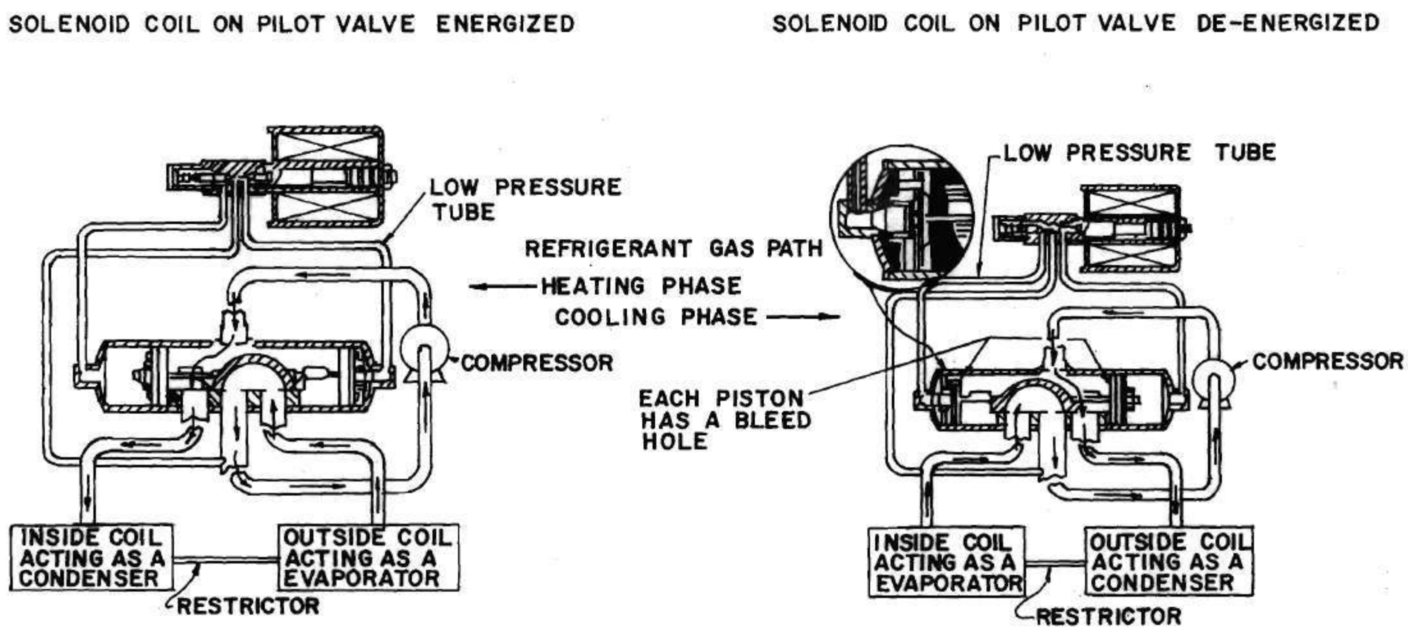


FIGURE 23.[9] REFRIGERANT GAS PATH IN REVERSING VALVE

Working Fluid

Let us consider, for a moment, the working fluid in the system under study. A refrigerant is a medium which absorbs heat by evaporating at a low temperature and gives up heat by condensing at a high temperature and pressure. An ideal refrigerant does not exist. Therefore, the particular fluid chosen for a specific application must be a compromise. It should have, however, as many of the following desirable characteristics as practicable:

(1) The latent heat of vaporization should be high. The higher this value, the greater will be the heat absorbed per pound of circulated fluid.

(2) The specific heat of the liquid should be low. The lower the specific heat of the liquid, the less heat it will pick up for a given change in temperature during either throttling or in flow through the piping and consequently the greater the refrigerating effect per pound of refrigerant.

(3) The specific volume should be low to minimize the work required per pound of refrigerant circulated.

(4) The critical temperature of the refrigerant should be higher than the condensing pressure to prevent excessive power consumption.

(5) The working fluid should have a boiling point below room temperature at atmospheric pressure and a freezing temperature well below any temperature at which the evaporator might operate.

In addition to the foregoing, it is necessary to consider items such as toxicity, flammability, explosiveness, corrosiveness, dielectric strength for a hermetically sealed system, viscosity, thermal conductivity, chemical stability, and cost. Once again, no one refrigerant has been found to meet all of these requirements.

Freon-22 or Refrigerant-22 (monochlorodifluoromethane, $CHClF_2$), a few properties of which are listed in Table 8, was used in the system analyzed in this work.

Table 8. Physical and Thermal Properties of Freon 22 [10]

Boiling Point:	-41.4°F
Freezing Point:	-256.0°F
Critical Point Temperature:	204.8°F
Critical Point Pressure:	716.0 PSIA
Specific Gravity of Liquid at Atmospheric Pressure:	1.411
Specific Heat of Liquid, Ave. 5°F to 86°F:	0.30

Freon 22 is used in a multitude of household and commercial applications. It is nontoxic and has a low power requirement per ton, and is a good low-temperature refrigerant, especially in applications utilizing reciprocating or rotary compressors. In recent years it has become the most popular refrigerant for residental air conditioning units, because its low boiling point and high latent heat permit the use of smaller compressors and refrigerant lines, making it ideal for compact units. Freon 22 is a good refrigerant for extreme service conditions because it is stable and has unusually good thermodynamic properties, equally true whether high capacity or high temperature stability is the major consideration.

Instrumentation

The parameters of interest included cycle pressures, cycle temperatures, the freon flow rate, and the electrical input. The cycle pressures (pressures measured before and after the four main system components) were measured on two pressure gauges. One was a Solfrunt Pressure Gauge (accuracy of about ±2%). The other, a Heise gauge (accuracy of about ±1%), was used to measure pressures greater than 300 PSIG, the range of the first gauge employed. The cycle temperatures (once again, the temperatures before and after the four main system components) were measured with iron-constantan thermocouples attached to the surface of the system lines, which were then insulated. The temperatures in (°F) were read directly off an Omega Engineering Inc. pyrometer (accuracy of approximately ±2%). A Rockwell flow meter allowed measurement of the liquid freon flow rate, while a General Electric, single-phase watthour meter, type 1-50-S, was used to determine the electrical input.

A box, open on both ends, was constructed of masonite and attached to the machine over the indoor coil side. In this box was built a movable baffle, which could be fixed at various positions. When horizontal, it, in effect, had no influence on the air flow rates over the indoor coil and hence, these cycle temperatures. When vertical, however, the exhaust air was effectively trapped and recirculated back into the machine by the indoor fan intake. This, as well as positioning of the baffle at various angles between the complete horizontal and the complete vertical positions, allowed some control over the simulated indoor temperatures. Unfortunately, this technique could not be used on the outdoor coil, due to the machine configuration. Instead, the air flows, and hence the temperatures, were controlled by covering the air passageways with plastic. These modifications allowed some control over the simulated indoor and outdoor temperatures. The obtainable range of conditions, however, was not the most desirable or the most comprehensive.

The experimental data, as well as sample calculations, for both the cooling and heating modes are presented in Appendix C.

Test Results

The experimental results for the electrical cooling cycle are tabulated below where:

T_L = the low (fluid evaporating) temperature

T_H = the high (fluid condensing) temperature

Capacity (tons) = (enthalpy change across the evaporator, Btu/#m) X (the mass flow rate, #m/hr)

$$COPW_c(\text{ideal}) = \frac{T_L/T_H}{1-T_L/T_H} \tag{5}$$

$COPW_c$(actual) = (enthalpy change across the evaporator, Btu/#m) X (the mass flow rate, #m/hr) ÷ $\dot{E}$ (the electrical input rate, Btu/hr)

Table 9. Electrical Heat Pump (Cooling) Test Results

Test	T_L(°F)	T_H(°F)	Capacity (tons)	$COPW_c$(actual)	$COPW_c$(ideal)
1	45.5	136.0	4.40	1.97	5.67
2	26.0	130.0	2.48	1.38	4.56
3	12.5	122.5	2.01	1.22	4.26
4	46.5	153.0	4.44	1.78	4.88
5	26.0	149.5	2.24	1.25	4.00
6	10.5	134.0	1.85	1.13	3.76
7	19.5	139.5	1.96	1.17	4.00
8	27.5	150.5	2.13	1.12	4.00

The heating cycle electric heat pump results are presented below.

T_L = the low (fluid evaporating) temperature

T_H = the high (fluid condensing) temperature

Capacity (Btu/hr) = (enthalpy change across the condenser, Btu/#m) X (the mass flow rate, #m/hr)

$$COPW_H(\text{ideal}) = \frac{1}{1 - T_L/T_H} \tag{6}$$

$COPW_H$(actual) = (enthalpy change across the condenser, Btu/#m) X (the mass flow rate, #m/hr) ÷ $\dot{E}$ (the electrical input rate, Btu/hr)

Table 10. Electrical Heat Pump (Heating) Test Results

Test	T_L(°F)	T_H(°F)	Capacity (Btu/hr)	$COPW_H$(actual)	$COPW_H$(ideal)
1	60.5	161.5	7.66×10^4	2.25	6.25
2	57.0	154.0	8.89×10^4	2.41	6.25
3	53.0	149.0	7.88×10^4	2.14	6.25
4	42.0	138.0	5.59×10^4	1.80	6.25

A comparison of the various systems is given in Chapter VI.

CHAPTER V

NATURAL GAS HEAT PUMP STUDY

The Wankel Engine as a Stationary Prime Mover

If the required work input to a heat pump system is provided by a natural gas-fueled engine, a natural gas heat pump results. This system has advantages over both a gas furnace and an electric heat pump. The advantage of the natural gas heat pump over a gas furnace results from the "free" heat which the evaporator absorbs from the outside air. Operating the compressor with a natural gas-fueled engine allows the waste heat accompanying the engine to be recovered and also used for residential heating. A significant efficiency advantage over an electric heat pump results from this waste heat recovery. In the electric system, the waste heat is generated at the power plant, but cannot be used, resulting in thermal pollution and a lowering of efficiency.

The primary problem to be solved in order to put the natural gas heat pump idea into commercial practice concerns the prime mover. One type of engine that shows promise for high reliability in this stationary application is the Wankel rotary engine. The Wankel has certain characteristics that render it superior to a conventional piston engine operating under similar conditions.

Operating on the Otto cycle, the Wankel engine has the same processes as a conventional reciprocating internal combustion engine. It is extremely simple in design, though, having basically only two moving parts, the rotor and the eccentric shaft (see Figure 24). The rotor revolves directly on the eccentric shaft eliminating the need for connecting rods. Also, valves and their operating mechanisms are not required since intake and exhaust gases pass through ports. Figure 25 shows an exploded view of the principle engine parts. The output torque is transmitted to the shaft through the eccentric. The internal and external gears shown are timing gears designed to maintain the phase relationship between the rotor and the eccentric shaft rotation. The smaller external gear (coaxial with the eccentric shaft) is fixed to one side of the housing [1].

Since the rotor is three-sided there are three separate cycles operating simultaneously. The processes which occur in sequence are shown in Figure 25. For each revolution of the rotor, there are three power impulses. Since the eccentric shaft rotates at three times the rotor speed, however, there is only one power pulse per rotor for each output shaft revolution [1].

The use of rotary engines in stationary applications is very attractive because of various advantages resulting from the simplicity of the Wankel design as compared to

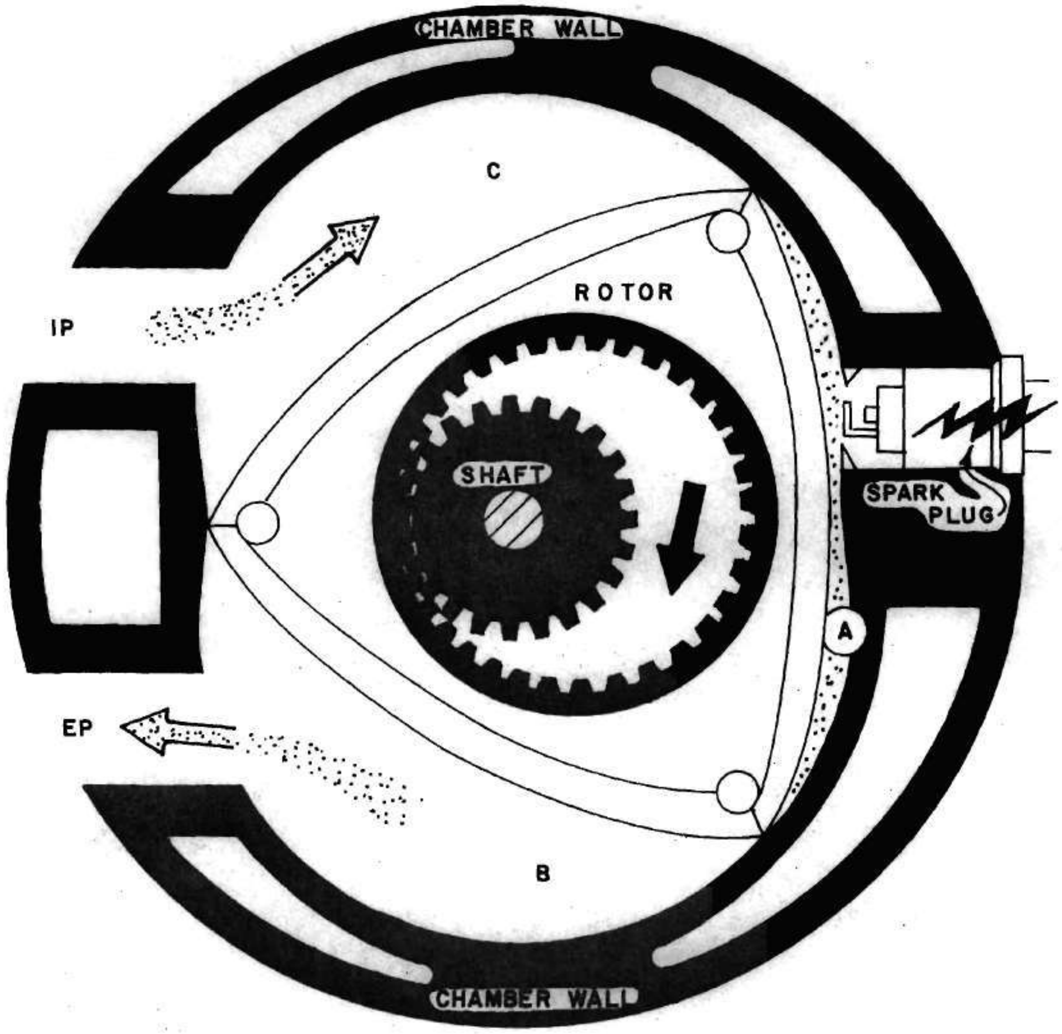


FIGURE 24.[11] ROTOR AND HOUSING OF A BASIC WANKEL ENGINE
A: COMPRESSION
B: POWER AND EXHAUST
C: INTAKE
IP = INTAKE PORT
EP = EXHAUST PORT

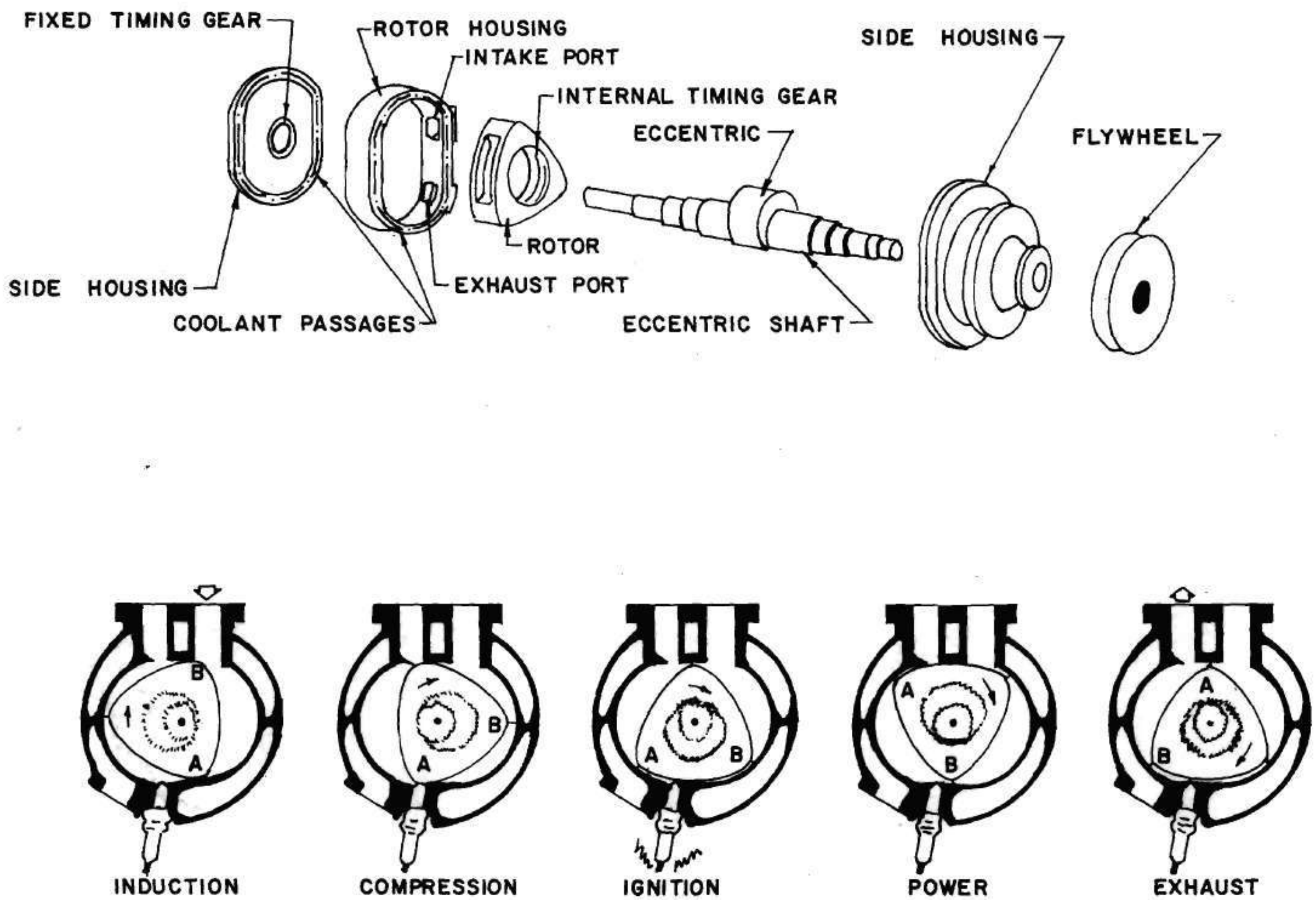


FIGURE 25.[1] BASICS OF A SINGLE ROTOR WANKEL ENGINE

conventional engines. Compared with a piston engine of comparable power output, the Wankel engine is about half the size and weight. Regardless of maximum power output, the Wankel engine has a far superior weight per horsepower ratio when compared with reciprocating engines [12]. Another major advantage is a reduction of parts, resulting in a far less complicated engine and providing greater simplicity and lower cost. Fewer moving parts also indicates less power loss due to internal friction. Finally, the small number of moving parts results in low operational costs since the required maintenance is limited, and the mechanical reliability is higher, thus extending the service life. It has been shown that the amplitude of vibration of a piston engine is over five times that of a Wankel [13]. This reduction of vibration is important in stationary long-life applications since a reduction in metal fatigue of various components results, as well as a dramatic reduction of noise levels. Finally, the Wankel engine encourages high volumetric efficiency because gas flow into and out of the combustion chamber is in a smooth sweep instead of through sharp turns or loops. In addition, compared with an intake stroke duration of 90 degrees of crankshaft rotation in a piston engine, the Wankel's suction intake occurs over 270 degrees of mainshaft rotation yielding a substantial volumetric efficiency advantage. Power output in a Wankel engine is smoother than in a piston engine because positive torque is produced for about two-thirds of the operating cycle as opposed to one-fourth or less of a four-stroke piston engine's cycle [12].

From a wear standpoint, the most critical components of the Wankel engine are the rotor apex seals. These seals, located at the apex of each rotor where they act as gas seals, have a rather short life when the Wankel is operated with gasoline. Tests conducted at Georgia Tech, however, indicate a much longer seal life when the engine is using natural gas [1]. This increased seal life would appear to give the Wankel the long-life characteristics needed in a natural gas heat pump.

Waste heat recovery of the prime mover is another area worthy of analysis in natural gas heat pumps. If the waste heat is of a low temperature, generally in the form of 150°F to 250°F cooling air or water and 1,000°F exhaust gases, the recovery equipment required is large and the system's capital cost is consequently increased. In reciprocating I.C. engines, the waste heat is usually split such that about 60% (or more) of the waste is contained in the cooling medium and 40% (or less) is in the exhaust gases [1].

In a Wankel engine the waste heat situation is somewhat different. Due to the short exhaust passages from the combustion chamber to the exhaust manifold, little exhaust cooling can occur and temperatures in the 1500°F to 2000°F range result. The exhaust gases at these high temperatures contain about 60% of the exhaust heat [1]. Thus, a majority of the Wankel engine's waste heat is at a high temperature making it easy to recover, resulting in smaller and less expensive heat recovery equipment.

Equipment and Instrumentation

The natural gas heat pump studied was the same electric system analyzed previously, with one major modification. The electric compressor was disconnected, and in its place, a Trane Model G, (G7H20X1), two-cylinder, reciprocating compressor driven by a Wankel engine was installed. Table 11 contains capacity data supplied by the compressor manufacturer [14].

A modification was also required on the compressor itself. Under the desired operating conditions it was determined that the required horsepower necessary to power the compressor and the capacity supplied were approximately double the values expected and desired. The problem was solved by removing one of the two pistons. The connecting rod was cut and a piston removed. The bearing and bearing cap were reattached to the crankshaft with special precautions taken to ensure against the bearing-turning or striking the compressor housing as the crankshaft rotated. Thus, problems of misbalance and improper oil lubrication were eliminated.

The engine used to power the system was a Sachs Wankel Engine Model KM914 A, single rotor, air-cooled

Table 11. Two-Cylinder Model G Compressor (3450 RPM) Performance Data [14]
Condensing Temperatures °F

Suction	110		120		130		140		150	
Temp °F	Tons	BHP	Tons	BHP	Tons	BHP	Tons	BHP	Tons	BHP
20	5.40	11.05	4.85	11.35	4.30	11.45	3.75	11.35	3.20	11.35
30	7.00	11.90	6.35	12.25	5.70	12.50	5.05	12.55	4.40	12.55
40	8.85	12.70	8.15	13.15	7.40	13.45	6.60	13.65	5.85	13.75
50	11.05	13.50	10.20	13.90	9.35	14.35	8.35	14.65	7.55	14.85

engine, a few specifications of which are given in Table 12. A Chrysler automotive solid state induction triggered (pointless) ignition system was adapted to the engine. Previous studies had been done on this engine at Georgia Tech [1]. Under the operating conditions of 3300 RPM, reference [1] predicts that the engine running on natural gas develops about eight horsepower and has a thermal efficiency of approximately 20%.

Waste heat recovery was achieved in the following manner. The Amana unit tested throughout this work was mounted approximately 36 inches above the floor on a frame built of 2-1/4 inch slotted angle material. In this part of the testing the Trane compressor and Wankel engine were placed on the floor beneath the Amana unit. See Figures 26, 27, 28, and 29 for system schematics. A stainless steel 1-1/2 inch exhaust pipe was installed between the engine exhaust and the Heat Transfer Module used in the gas furnace analysis. This module and the water pump and circulatory system were used in the exhaust heat recovery process. All unnecessary components in the gas furnace apparatus were disconnected and removed at this stage. In this testing no insulation was placed on the exhaust pipe. It was felt that the effort necessary to find, obtain, and install insulation capable of withstanding the 1,500-2,000°F temperatures was not worthwhile since the installed insulation would probably not be as effective in preventing heat loss as insulation in

Table 12. KM914 Engine Specifications [12]

Displacement:	18.5 cu. in. (303 cc)
Cooling:	Air-cooled (integral blower)
Compression ratio:	8:1
Performance (Gasoline):	20.0 horsepower at 5,000 RPM (tolerance range +5%)
Ignition timing (gasoline):	10°...12° before top dead center
Contact breaker gap:	0.014-0.018 in
Carburetor:	Tillotson diaphram carburetor HL-268A 22 mm (0.866 in) dia. or Bing butterfly valve carburetor 22 mm (0.866 in) dia.
Air filter:	Inlet mesh filter
Weight:	56 lbs., engine including starter, carburetor, and muffler
Pre-mix ratio:	40:1 fuel to oil

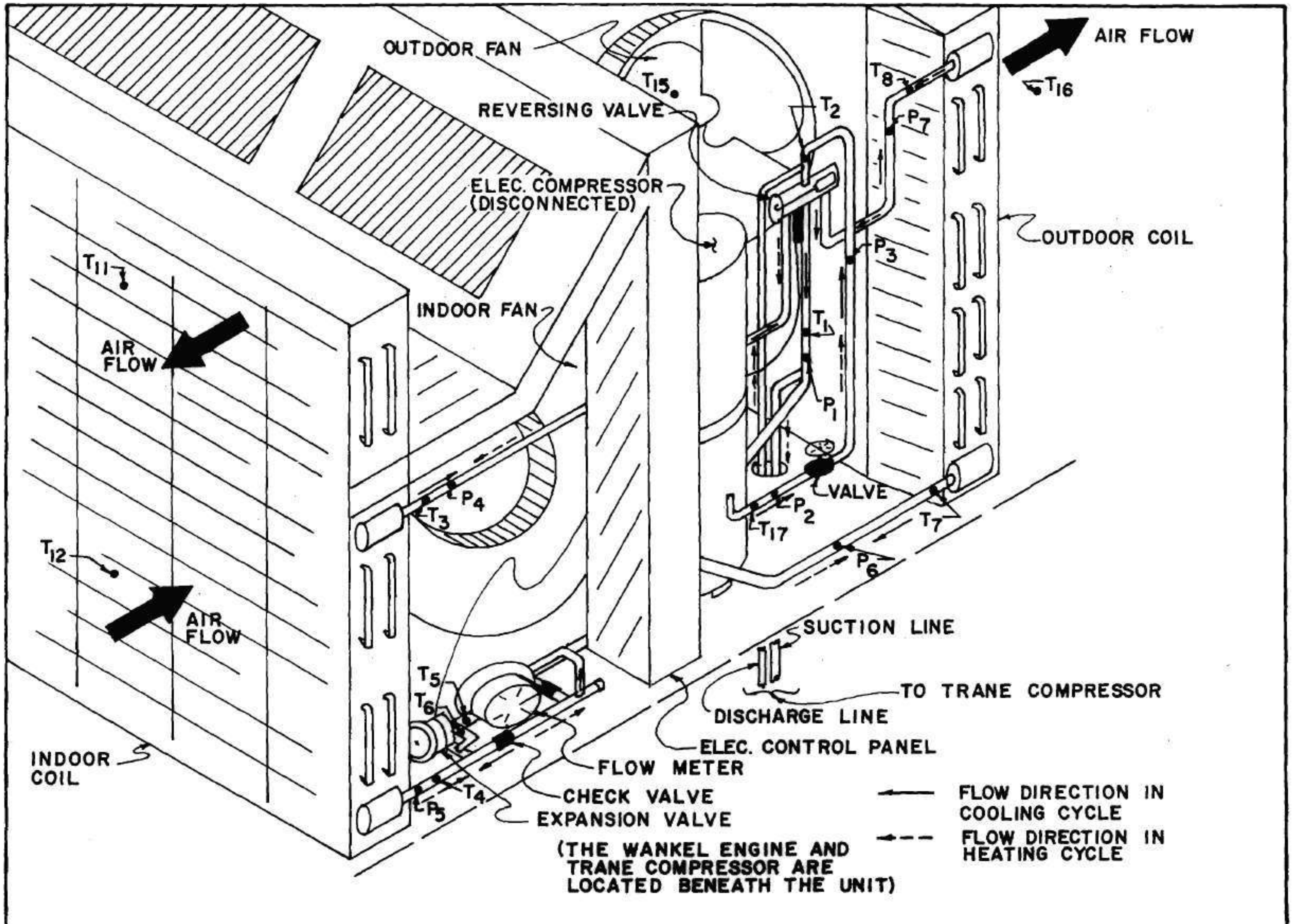


FIGURE 26. GAS HEAT PUMP SYSTEM SCHEMATIC INDICATING THE LOCATION OF MEASUREMENT POINTS

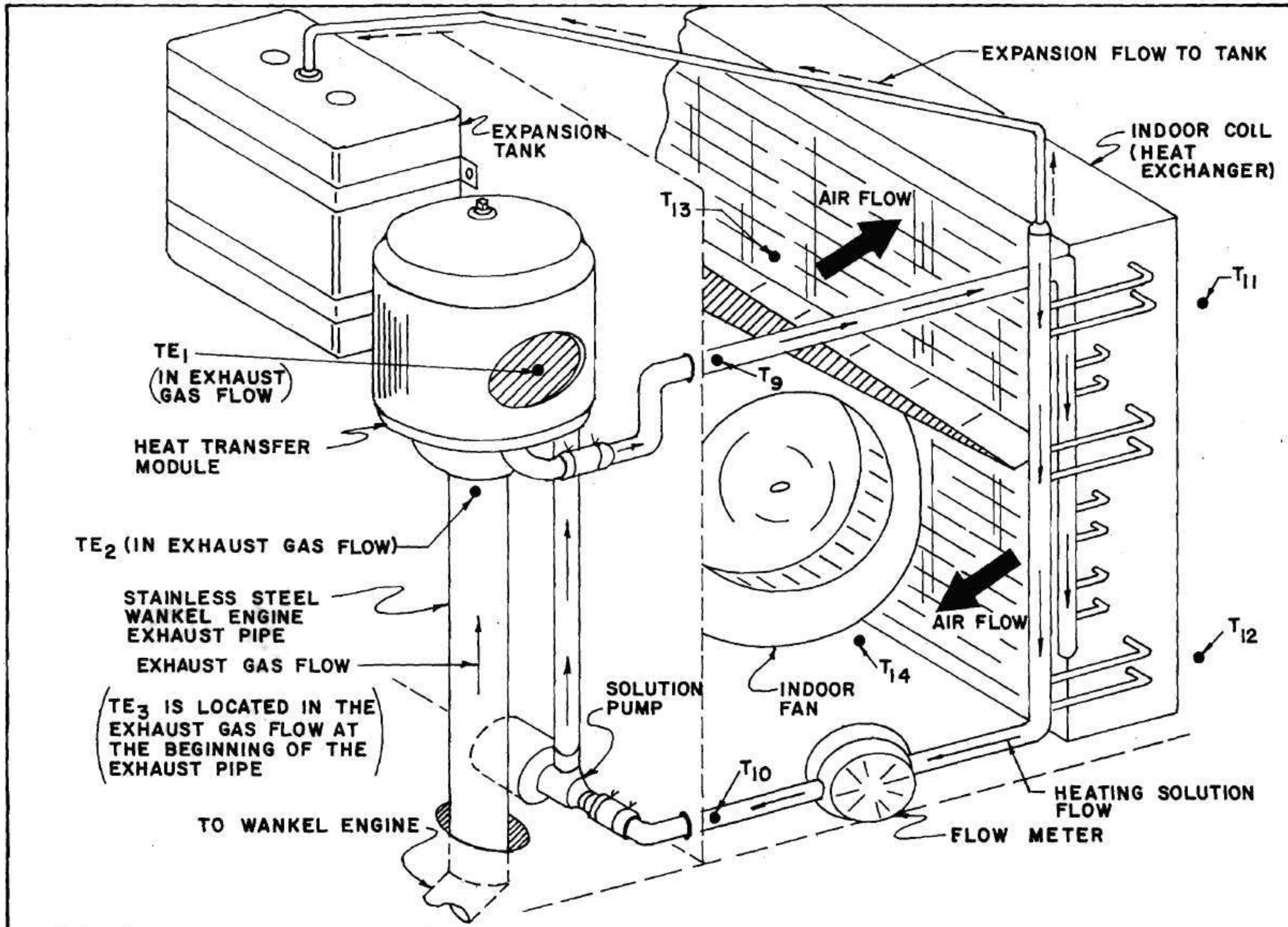


FIGURE 27. GAS HEAT PUMP WASTE HEAT RECOVERY SYSTEM SCHEMATIC INDICATING THE LOCATION OF MEASUREMENT POINTS

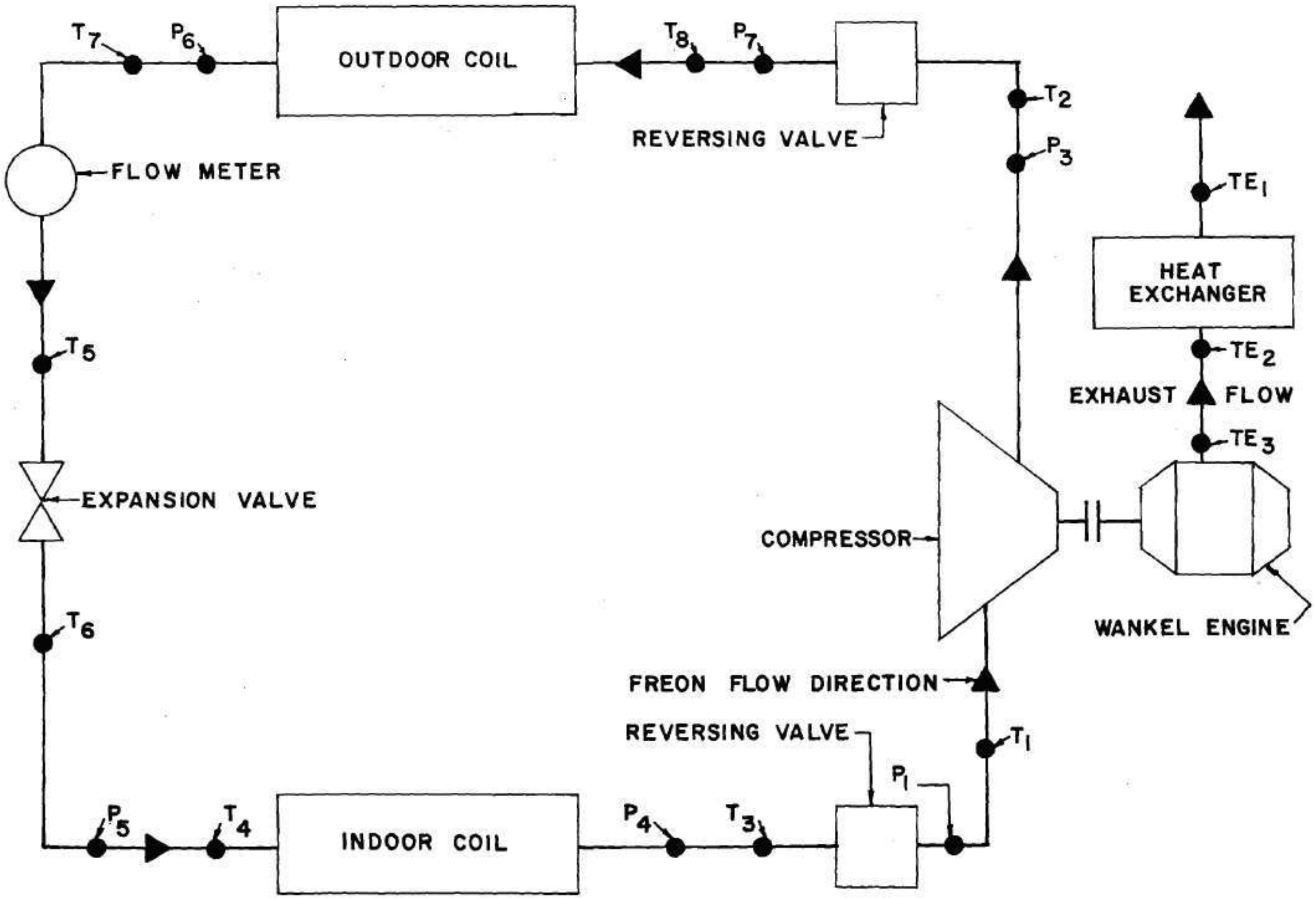


FIGURE 28. SCHEMATIC OF THE GAS HEAT PUMP COOLING CYCLE INDICATING THE LOCATION OF THE SIGNIFICANT MEASUREMENT POINTS

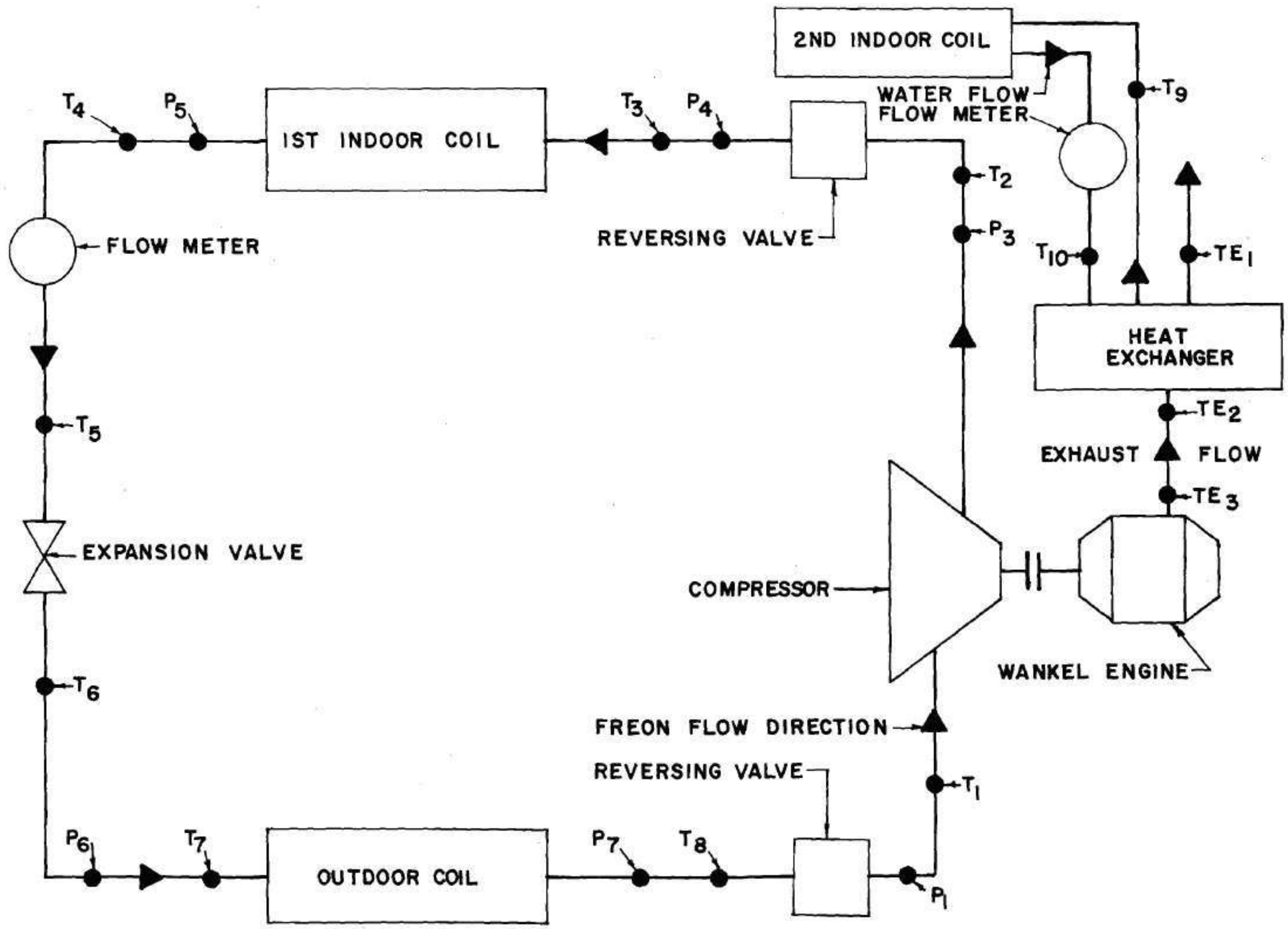


FIGURE 29. SCHEMATIC OF THE GAS HEAT PUMP HEATING CYCLE INDICATING THE LOCATION OF THE SIGNIFICANT MEASUREMENT POINTS

a commercially produced unit. Instead, three thermocouples were installed in the exhaust flow: one at the engine exhaust, a second 40 inches away, just before entry into the heat exchanger (Heat Transfer Module), and a third in the exhaust flow just after this unit, as the exhaust is about to be discharged to the atmosphere. Thus, the actual heat loss could be determined and an analysis is included predicting the possible performance if suitable insulation were installed. Insulation is recommended, of course, for any operating unit of this type.

The instrumentation of this system was similar to that previously described. The cycle temperatures and pressures, freon flow rate, and electrical input were measured with the same equipment described previously in the section on the electric heat pump. The waste heat recovery was accomplished with the water system of the natural gas furnace originally studied. Thus, the water and gas flow rates and appropriate water temperatures were obtained using the instruments described earlier in the gas furnace study. The only additional measurement made was of exhaust gas temperatures. These were made using chromel-alumel thermocouples inserted into the exhaust gas flow. The temperatures were read on an Omega type 2809 digital thermometer (accuracy = ±1%).

Test Results

The experimental data, as well as sample calculations, for both the cooling and heating modes, is presented in Appendix D. The following results were obtained:

T_L = the low (fluid evaporating) temperature

T_H = the high (fluid condensing) temperature

Capacity (tons) = (enthalpy change across the evaporator, Btu/#m) X (the mass flow rate, #m/hr)

$$COPH_c\text{(ideal)} = \frac{T_L/T_H - T_H/T_S}{1 - T_L/T_H} \text{ ; (assume } T_S = 1{,}000°\text{F)} \tag{14}$$

$COPH_c$(actual) = (enthalpy change across the evaporator, Btu/#m) X (the mass flow rate, #m/hr) ÷ $\dot{G}$ (the gas input rate, Btu/hr)

Table 13. Gas Heat Pump (Cooling) Test Results

Test	T_L(°F)	T_H(°F)	Capacity (tons)	$COPH_c$(actual)	$COPH_c$(ideal)
1	60.0	142.5	6.80	.674	3.21
2	61.5	141.5	11.42	1.04	3.54
3	65.0	154.25	9.73	.898	2.87

Similarly for the heating cycle:

T_L = the low (fluid evaporating) temperature

T_H = the high (fluid condensing) temperature

Capacity (Btu/hr) = (enthalpy change across the condenser, Btu/#m) X (the mass flow rate, #m/hr)

$$COPH_H(\text{ideal}) = \frac{1-T_H/T_S}{1-T_L/T_H}\ ;\ (\text{assume } T_S = 1{,}000°F) \quad (15)$$

$COPH_H$ (actual) = (enthalpy change across the condenser, Btu/#m) X (the mass flow rate, #m/hr) ÷ $\dot{G}$ (the gas input rate, Btu/hr)

Refer to Table 14 where two of the sets of values include waste heat recovery, $\dot{Q}_{rec}$.

$$\dot{Q}_{rec} = \dot{m}_w \, \Delta T_w \, C_{pw} \quad (47)$$

where:

$\dot{m}_w$ = the cooling water mass flow rate, #m/hr

ΔT_w = the water temperature change across the heat exchanger, °F

C_{pw} = water specific heat, 1 Btu/(#m°F)

At the engine exhaust the measured temperature was 1770°F. After 40 inches of uninsulated stainless steel exhaust pipe, the measured temperature was 1175°F. Finally, the exhaust passes through the Heat Transfer Module and upon exiting the temperature dropped to 170°F. (These measured values varied somewhat among the various tests, but the particular temperatures used in this calculation were chosen to give an indication of the improvement that could possibly be achieved. Thus, the maximum engine exhaust temperature measured was used.)

Table 14. Gas Heat Pump (Heating) Test Results

Test	T_L(°F)	T_H(°F)	Capacity (Btu/hr)	$COPH_H$(actual)*	$COPH_H$**	$COPH_H$***	$COPH_H$(ideal)*
1	73.0	133.0	9.42×10^4	0.83	1.11	1.29	5.90
2	81.5	137.75	8.10×10^4	0.78	1.09	1.18	6.56
3	78.25	139.5	9.26×10^4	0.87	1.18	1.37	5.90
4	67.5	148.0	8.63×10^4	0.69	0.95	1.12	4.46

*With no waste heat recovery

**With actual waste heat recovery

***With waste heat recovery assuming an insulated exhaust

If a modified $COPH_H$ is now defined as:

$$COPH_H = \frac{\dot{Q}_{cond} + \dot{Q}_{rec}}{\dot{G}} \qquad (48)$$

where:

$\dot{Q}_{cond}$ is the numerator in the previously defined equations

$\dot{G}$ is the previously defined gas input

$\dot{Q}_{rec}$ is the waste heat recovery

the values of $COPH_H$ are significantly improved. Sample calculations are included in Appendix D.

An analysis can be made to determine the benefit of insulating the exhaust, thereby recovering more of the waste heat and increasing the $COPH_H$. Defining the waste heat recovery as follows:

$$\dot{Q}_{rec} = \dot{m}_w \int_{ex} C_{pw}\, dT = \eta_{ex}\, \dot{m}_{ex} \int_{htm} C_{pex}\, dT \qquad (49)$$

where:

$\dot{Q}_{rec}$ = the waste heat recovery rate

$\dot{m}_w$ = the mass flow rate of the heat recovery water

$\int_{ex} dT$ = integration with respect to the temperature change of the heat recovery water across the heat exchanger (indoor coil)

C_{pw} = the specific heat of the heat recovery water

η_{ex} = the efficiency of heat transfer from the exhaust gases to the heat recovery water

$\dot{m}_{ex}$ = the exhaust gas mass flow rate

$\int_{htm} dT$ = integration with respect to the temperature change of the exhaust gases across the heat transfer module

C_{pex} = the specific heat of the exhaust gases

The variation in specific heat was taken into account to provide greater accuracy.

Assuming a stoichiometric methane combustion reaction,

$$CH_4 + 2O_2 + 2(3.76)N_2 \rightarrow CO_2 + 2H_2O + 7.52N_2$$

and using Table A.7 in reference [15] for the products of combustion,

$$CO_2: \quad C_{po} = 16.2 - \frac{6.53x10^3}{T} + \frac{1.41x10^6}{T^2}$$

$$H_2O: \quad C_{po} = 19.86 - \frac{597}{\sqrt{T}} + \frac{7500}{T}$$

$$N_2: \quad C_{po} = 9.47 - \frac{3.47x10^3}{T} + \frac{1.16x10^6}{T^2}$$

a comparison between the actual heat recovery and the ideal insulated exhaust recovery that possibly could be achieved can be made.

$$\frac{\dot{Q}_{rec}\text{(insulated exhaust)}}{\dot{Q}_{rec}\text{(actual)}} = \frac{\eta_{ex}\ \dot{m}_{ex}\ \int_{630R}^{2230R} C_p\ dT}{\eta_{ex}\ \dot{m}_{ex}\ \int_{630R}^{1635R} C_p\ dT}$$

The enthalpy change (dh) of the mixture was calculated using:

$$dh_{mixture} = \frac{(M_{co_2}\ dh_{co_2} + M_{H_2O}dh_{H_2O} + M_{N_2}dh_{N_2})}{(M_{co_2} + M_{H_2O} + M_{N_2})} \qquad (50)$$

Sample calculations are given in Appendix D. The result is

$$\frac{\dot{Q}_{rec}\text{(insulated exhaust)}}{\dot{Q}_{rec}\text{(actual)}} = 1.65$$

Therefore, the $\dot{Q}_{rec}$ could be increased from 32,182.08 Btu/hr to 53,100.43 Btu/hr by exhaust pipe insulation and the $COPH_H$'s increased, as summarized once again in Table 15.

Table 15. Gas Heat Pump $COPH_H$'s

Test	(Without Waste Heat Recovery)	(With Actual Waste Heat Recovery)	(With Waste Heat Recovery Assuming an Insulated Exhaust)
1	0.83	1.11	1.29
2	0.78	1.09	1.18
3	0.87	1.18	1.37
4	0.69	0.95	1.12

A comparison of the various systems follows in Chapter VI.

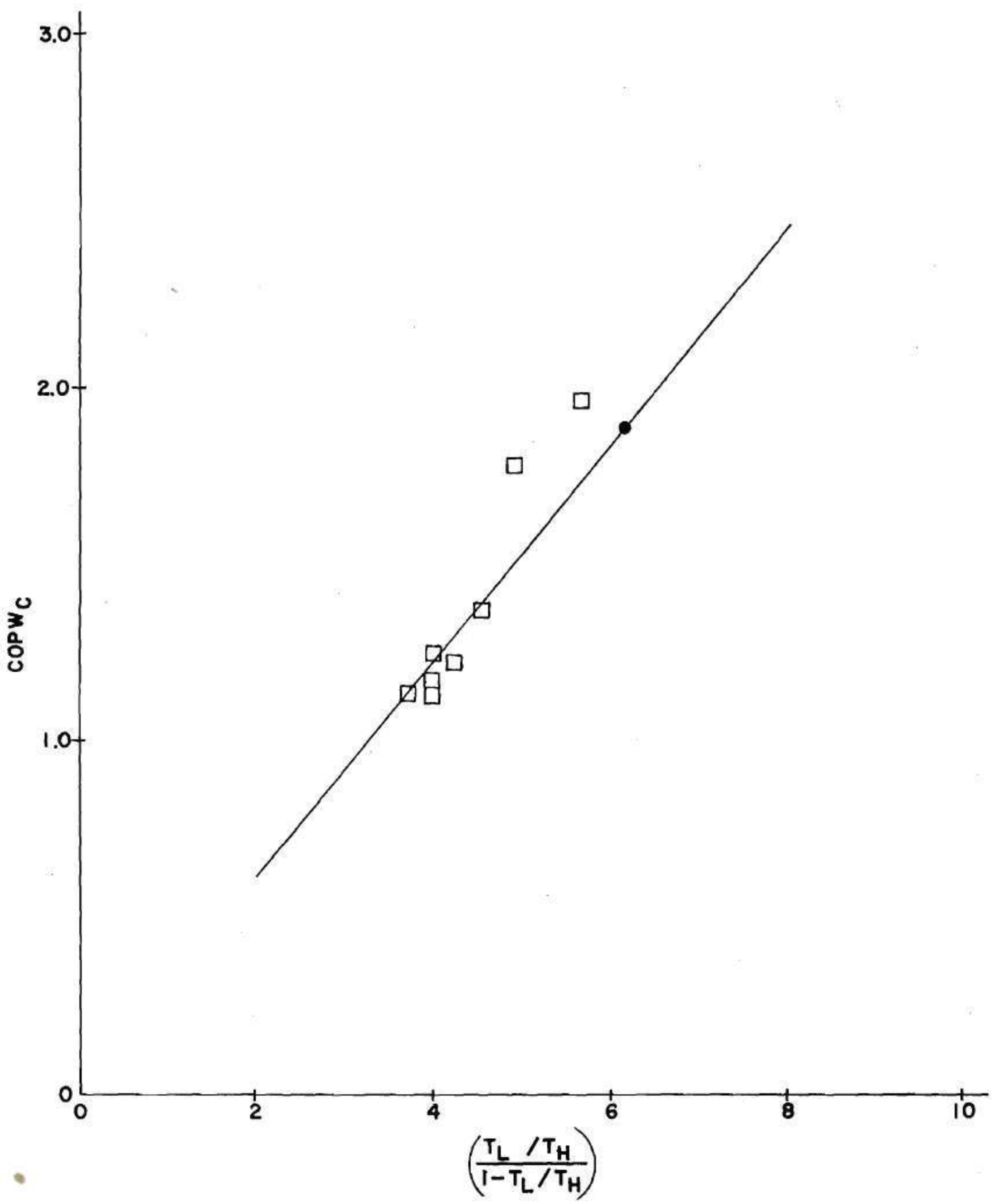
CHAPTER VI

DISCUSSION AND CONCLUSIONS

Comparison of Results

Let us compare, at this point, the test results of the various systems. Once again, relative to the electric heat pump system, an important point should be made. The COPH's desired in this study are defined as the cooling or heating output relative to the heating value of the fuel at the power plant. For an electrical system, the electrical output to the house is generally taken as 0.3 of the fuel input at the power plant, making the electrical system COPH's the product of the experimentally measured values (COPW's) and 0.3 [1].

Earlier in this work a relationship was established between the Carnot or ideal COPW's and the ideal vapor-compression cycle COP's. (See Figures 11, 12, 13, and 14.) At this stage, similar graphs, (Figures 30, 31, 32, and 33), can be formulated using the actual experimentally determined test cycles instead of ideal vapor-compression cycles. Refer to Appendix A for sample calculations and refer to the Vapor-Compression Cycle Analysis section for a brief discussion of K_1 and K_2. A comparison of the various K_1's and K_2's for the systems indicates the degree to which the actual cycles



□ EXPERIMENTAL VALUES; ● $T_L = 40°F$ & $T_H = 120°F$

FIGURE 30. ELECTRIC COOLING COPW$_C$ VS. $\left(\frac{T_L / T_H}{1 - T_L / T_H}\right)$; SLOPE = K_1 = .31

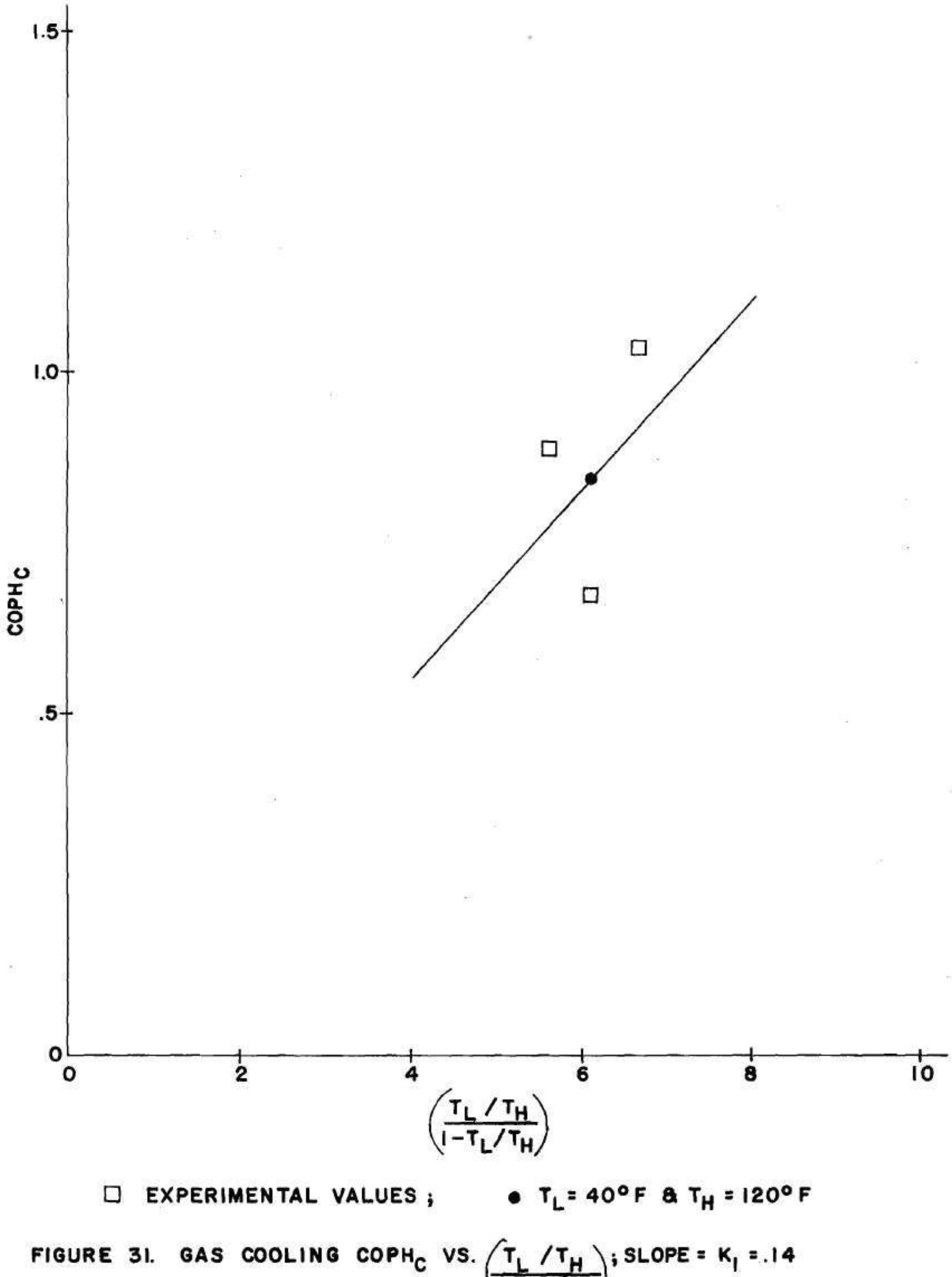
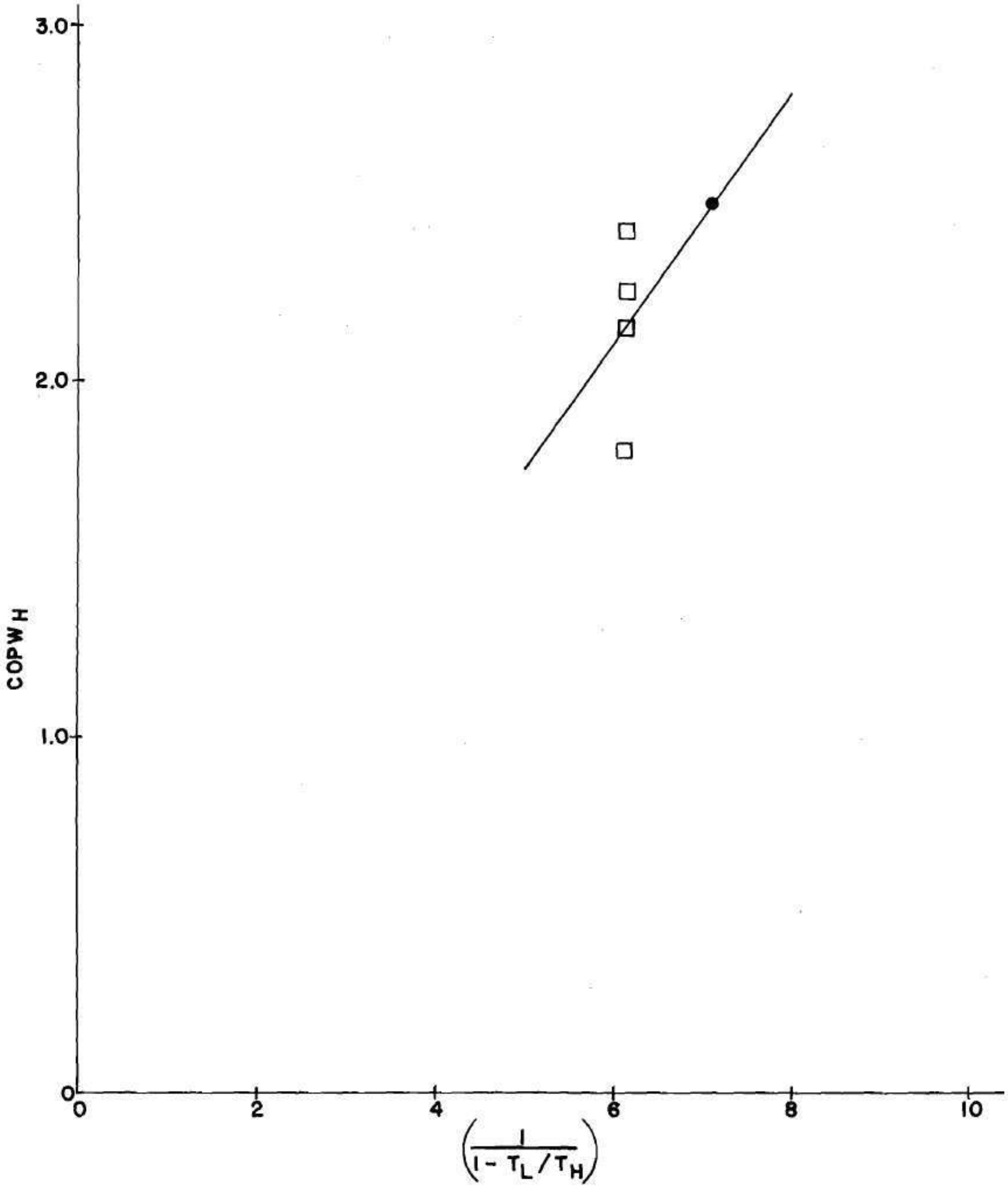
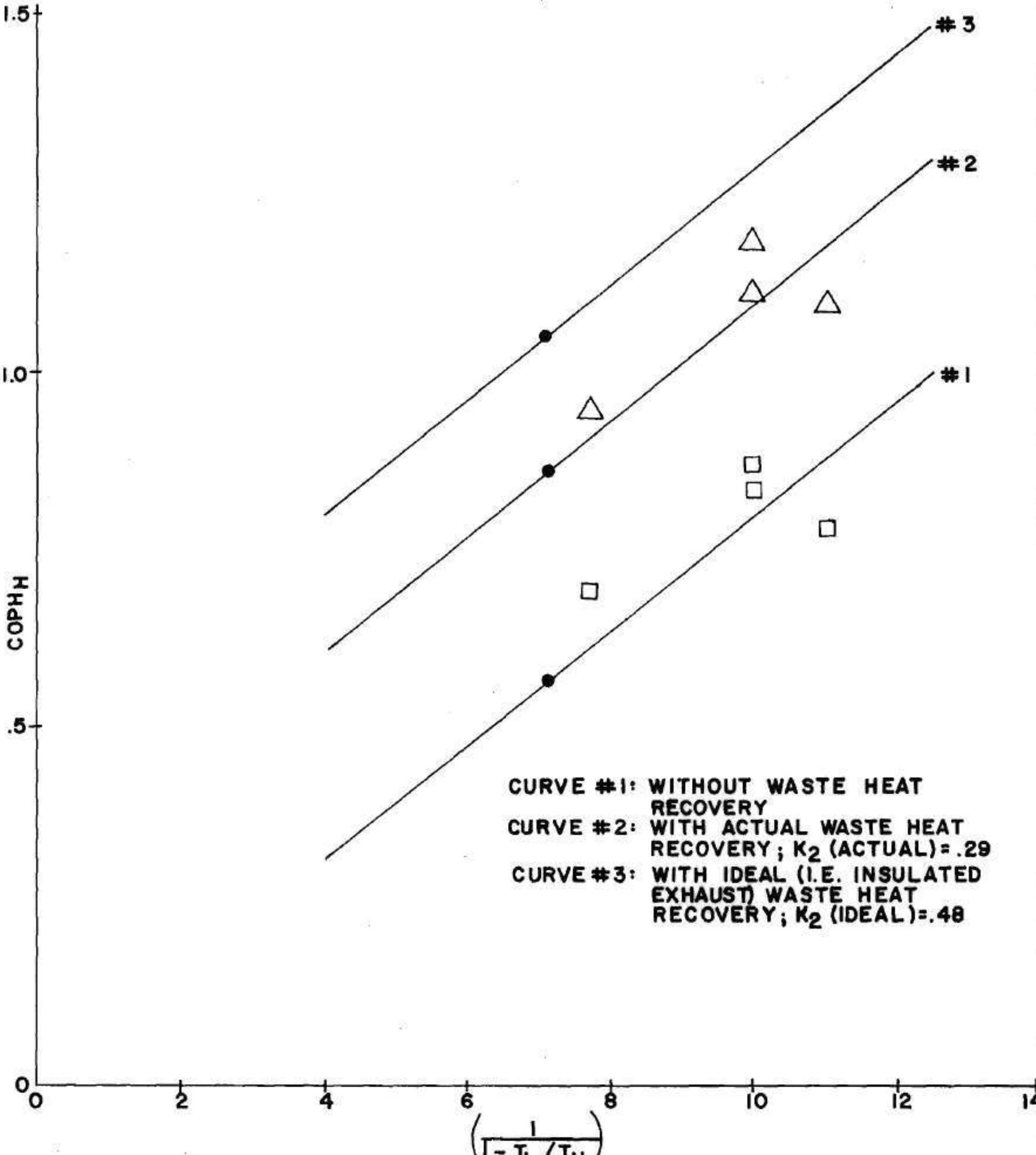


FIGURE 31. GAS COOLING COPH$_C$ VS. $\left(\frac{T_L / T_H}{1 - T_L / T_H}\right)$; SLOPE = K_1 = .14



□ EXPERIMENTAL VALUES; • $T_L = 40°F$ & $T_H = 120°F$

FIGURE 32. ELECTRIC HEATING $COPW_H$ VS. $\left(\frac{1}{1-T_L/T_H}\right)$; SLOPE = K_1 = .35



□ REPRESENT ACTUAL TEST RESULTS WITHOUT WASTE HEAT RECOVERY

△ REPRESENT ACTUAL TEST RESULTS WITH ACTUAL WASTE HEAT RECOVERY

● REPRESENT $T_L = 40°F$ & $T_H = 120°F$

FIGURE 33. GAS HEATING COPH$_H$ VS. $\left(\frac{1}{1 - T_L/T_H}\right)$; SLOPE = K_1 = .08

approach the ideal.

Once again, $(\frac{T_L/T_H}{1-T_L/T_H})$ approaches 0.0 as a lower limit while $(\frac{1}{1-T_L/T_H})$ approaches 1.0. For the electrical system, (Figure 32), at this lower limit $COPW_H$ equals 1.0. In the gas system analysis, (Figure 33), at the lower limit $COPH_H$ will equal the actual engine efficiency for the case of no waste heat recovery. If waste heat recovery is included, $COPH_H$ approaches a value equivalent to the actual engine efficiency plus the percentage of waste heat recovered.

One motivation for these graphs was to solve the problem created by the actual test result's limited range of condenser and evaporator temperatures. By developing the relationships mentioned above, much broader, and hence, much more useful results can be presented. The performance of a particular system under conditions other than those established in this experimental work can thus be estimated.

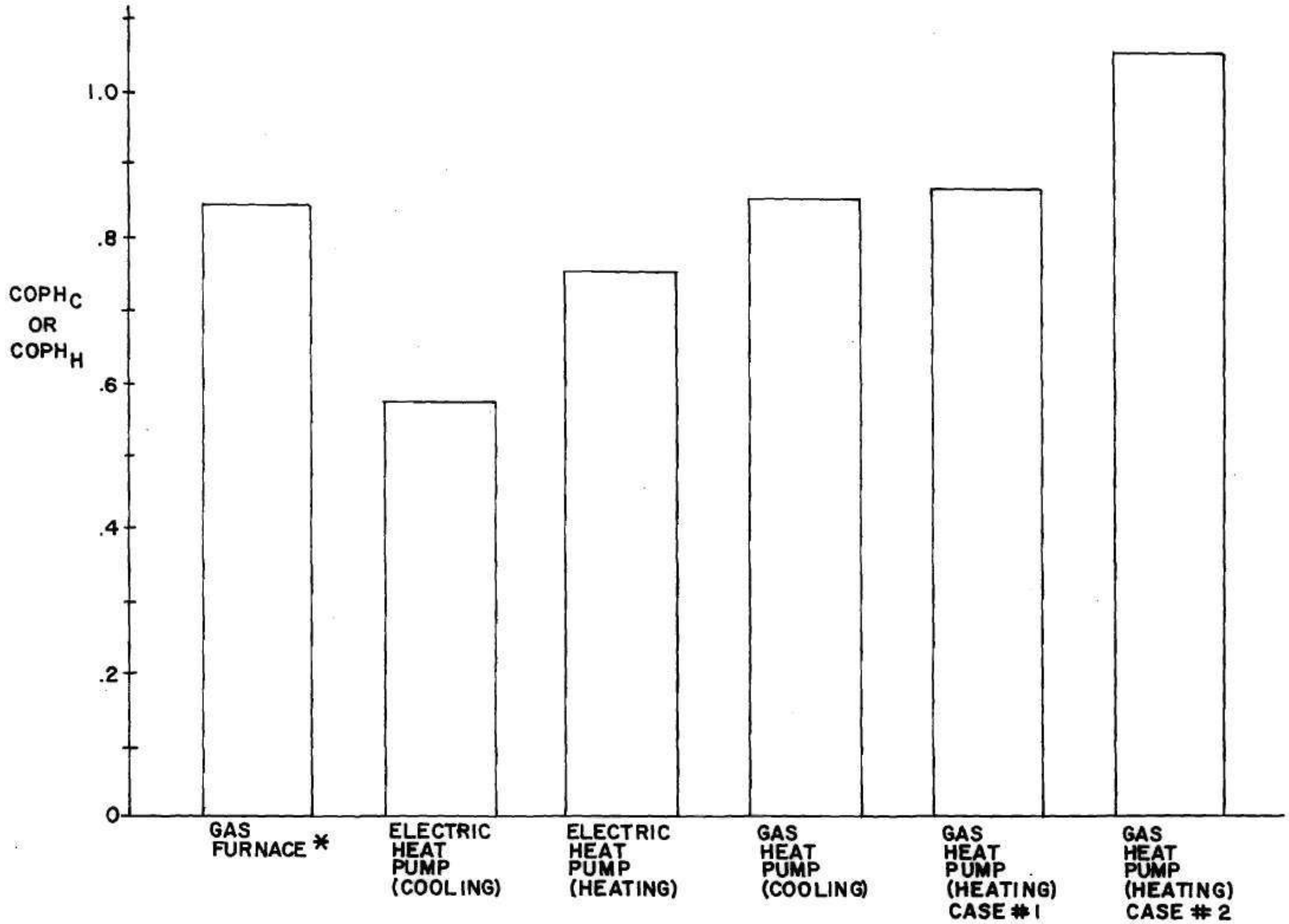
Perhaps the best method of comparing the alternate systems is to simply compare their performance at a given set of operating conditions. An analysis was performed for a representative condition of T_L = 40°F and T_H = 120°F, sample calculations of which are included in Appendix E. The results of this analysis are presented in Table 16 and Figures 34, 35, and 36.

The gas furnace data was included for completeness and somewhat limited comparison purposes. It should be recalled, however, that its performance is not dependent on

Table 16. Comparison of Alternate Systems (Based on performance with T_L = 40°F and T_H = 120°F)

System	$COPH_c$	$COPH_H$	$\$/10^6$Btu	Btu/$
Gas Furnace*	---	0.84	1.55	6.45×10^5
Electric Heat Pump (Cooling)	0.57	---	4.63	2.16×10^5
Electric Heat Pump (Heating)	---	0.75	2.34	4.27×10^5
Gas Heat Pump (Cooling)	0.85	---	1.29	7.75×10^5
Gas Heat Pump (Heating) (with actual waste heat recovery)	---	0.86	1.51	6.62×10^5
Gas Heat Pump (Heating) (with waste heat recovery assuming an insulated exhaust)	---	1.05	1.24	8.06×10^5

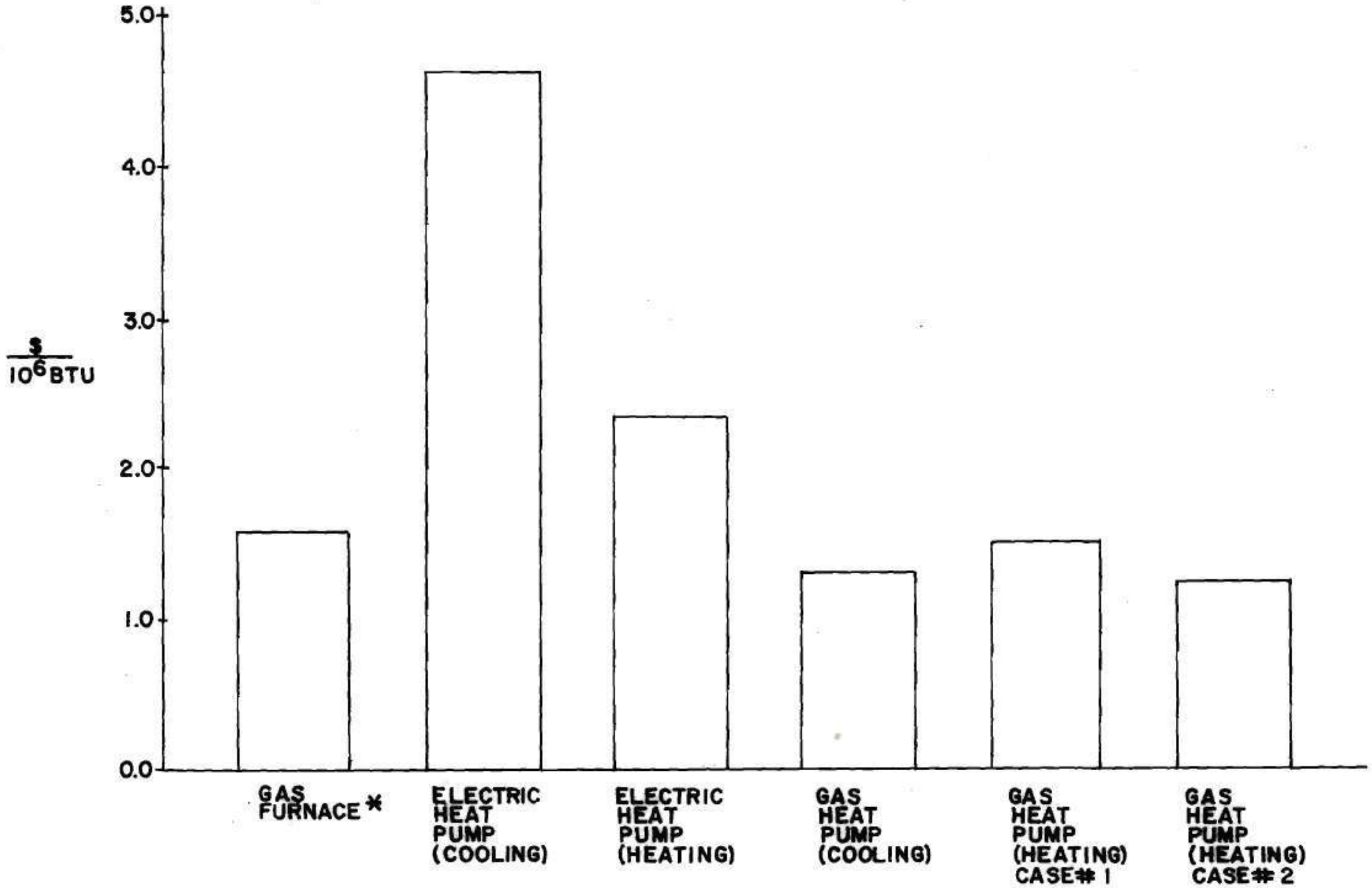
*The gas furnace values are simply averages of the measured results, which are not restricted to T_L = 40°F and T_H = 120°F.



CASE #1: WITH ACTUAL WASTE HEAT RECOVERY

CASE #2: WITH WASTE HEAT RECOVERY ASSUMING AN INSULATED EXHAUST

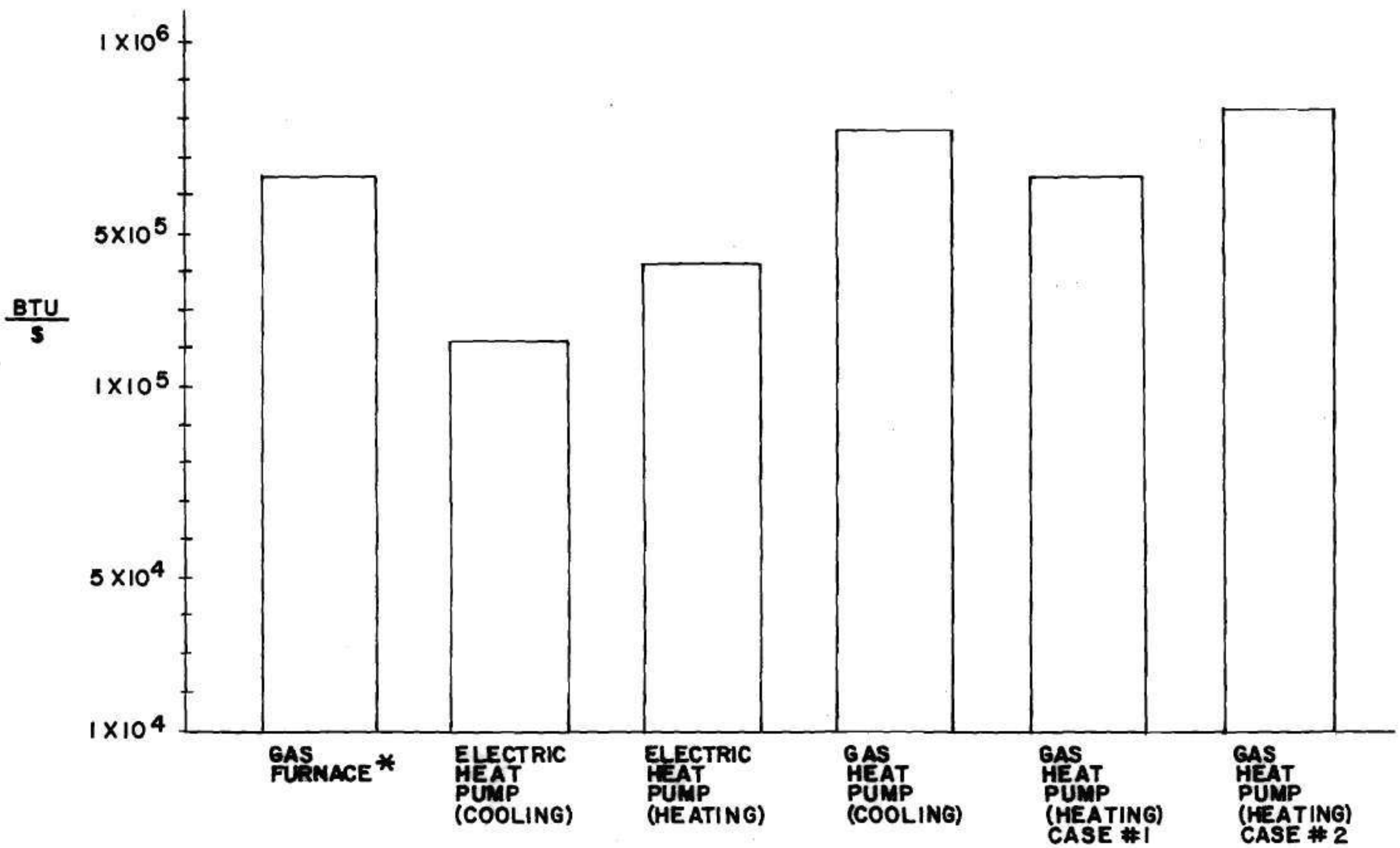
FIGURE 34. COMPARISON OF COOLING AND HEATING COPH'S FOR ALTERNATE SYSTEMS (T_L=40°F, T_H=120°F); *SEE TABLE 16.



CASE #1: WITH ACTUAL WASTE HEAT RECOVERY

CASE #2: WITH WASTE HEAT RECOVERY ASSUMING AN INSULATED EXHAUST

FIGURE 35. ($/$10^6$ BTU) COMPARISON OF ALTERNATE SYSTEMS (T_L = 40°F, T_H = 120°F); *SEE TABLE 16.



CASE #1: WITH ACTUAL WASTE HEAT RECOVERY

CASE #2: WITH WASTE HEAT RECOVERY ASSUMING AN INSULATED EXHAUST

FIGURE 36. (BTU/$) COMPARISON OF ALTERNATE SYSTEMS (T_L = 40°F, T_H = 120°F); *SEE TABLE 16.

T_L and T_H in the same manner as the heat pump systems. The values listed are simply the averages of its experimentally determined performance characteristics. Moreover, since it lacks a cooling mode, a true comparison between the gas furnace system and the other alternates cannot be made.

A comparison of COPH's at this particular operating condition shows the gas heat pump's COPH's to be significantly higher. In the cooling mode, the gas heat pump $COPH_C$ is about 49% greater than that of the electric heat pump. In the heating condition, the gas heat pump $COPH_H$ (with the actual waste heat recovery) is only about 2% greater than the gas furnace analyzed, but about 15% greater than the $COPH_H$ for the electric heat pump. If waste heat recovery assuming insulation is included, these values become approximately 25% greater than the gas furnace and 40% greater than the electrical system. Recall that the gas furnace analyzed here performed much better than a conventional gas furnace. Also, the electrical system values include allowance for power plant inefficiencies.

The consumer cost differential between the electric and gas systems would be even more favorable to the gas systems than these efficiency comparisons [1]. A comparison of $\$/10^6$ Btu and Btu/\$ for the alternate systems would perhaps be more meaningful than COPH comparisons. These values were calculated using the parameters listed below which were confirmed by the appropriate Atlanta utilities.

Current Typical Atlanta Residential Rates

	Heating (winter)	Cooling (summer)
Electric	\$.02/KW-hr	\$.03/KW-hr
Gas	\$1.30/10^6Btu	\$1.10/10^6Btu

The cost per 10^6 Btu comparison produces some astounding results. For 10^6 Btu of cooling, using the above cost factors, the electric system cost is about 259% greater than the gas heat pump system cost. In the heating mode, the electric system cost is approximately 55% greater than that of the gas heat pump with actual waste heat recovery, and 89% greater than the gas system with waste heat recovery assuming an insulated exhaust. Similarly, the gas furnace shows a heating cost 3% greater and 21% greater than the gas heat pump with actual and insulated exhaust waste heat recovery, respectively.

To briefly summarize these comparisons, it appears that a natural gas-fueled Wankel engine-driven heat pump system could offer a dramatic reduction in energy consumption for residental heating and cooling, the two largest residential energy usages. In addition, the possible cost savings that might be realized are truly impressive.

Conclusions

This thesis has presented the results of an experimental study of a gas furnace, an electric heat pump, and a natural gas heat pump. Since the ultimate objective is to find a system that provides year-around environmental control for a home, (i.e. both cooling and heating), the comparison is essentially between the two heat pump systems. Reference [1] predicted a higher COPH for both cooling and heating for a natural gas-fueled Wankel engine-driven heat pump than for a similar system driven by an electric motor. The results of this thesis support that prediction.

These results should not be a startling revelation. An electric heat pump is a heat-actuated system that uses a Rankine heat engine at the generating plant to drive a refrigeration cycle through an electric motor at the dwelling [2]. A natural gas heat pump, on the other hand, employs a heat engine to drive a refrigeration cycle right at the point of use. It seems more logical to convert a fuel to sensible heat at its point of use, rather than convert it to another form of energy at a remote generating station and then reconvert it to sensible heat. Any electric system suffers from power plant inefficiencies as well as transmission losses.

An analysis was made in reference [6] of energy losses in natural gas and electrical transmission. The results are as follows. For short distances, approximately 10 to 20 miles, an average value of the electrical transmission efficiency, the ratio of energy out to energy in, is 95%. The losses, a 2% transmission line loss (I^2R loss) and a 3% loss in transformers, are a function of the distribution system loading, which varies greatly during the course of a year. The energy losses in natural gas transmission pipelines, however, are much smaller. The average energy requirements for transporting natural gas short distances, approximately 100 miles, is approximately 2% of the energy delivered to the pipeline inlet. The transmission efficiency is therefore 98% for this case, while it is estimated to be 95% for longer distances.

Natural gas is one of our premium fossil fuels. It is clean burning, requires no refining, and is easily recovered and shipped with little or no detrimental environmental impact. A natural gas-fueled Wankel engine-driven heat pump can provide a very efficient method of home environmental control, thus satisfying a very important energy requirement, and also, conserving our natural gas resources. In addition, this system is quite compatible with the environment.

The possible consequences of this system are rather amazing. This experimental study supports the analysis by the American Gas Association showing that if natural gas heat pumps were available for installation beginning in 1976, the accumulative savings in natural gas would be about 12×10^{12} cubic feet by 1990 (or 3×10^{12} cubic feet annually) [1,16]. This is based on the assumption that only 40% of the raw heating unit market by 1990 will be made up of natural gas heat pumps. The quantity of gas saved would be equivalent to that projected to be produced from coal gasification at a capital expenditure of about $16 billion [1,16].

A crisis in energy is one of the most important problems that this country, and indeed the world, faces at present. Unfortunately, this problem will not vanish by itself in time. Wise utilization of our natural resources is no longer a desired goal. At this point, it is an absolute necessity, making research in this area, needless to say, extremely important. It is the author's hope that this thesis, which concerns a study of systems to provide residential heating and cooling, the two largest residential energy usages, may help, in some small way, that effort.

APPENDICES

APPENDIX A

SAMPLE CALCULATIONS FOR FIGURES 11, 12, 13, 14, 30, 31, 32, and 33

All thermodynamic properties were found in Freon 22 tables [17]. Refer to Figure 8 for the cycle considered in this analysis and the state point locations.

Assume $T_{evap} = 20°F$ and $T_{cond} = 80°F$; $\frac{T_L/T_H}{1-T_L/T_H} = 8.09$;

$\frac{1}{1-T_L/T_H} = 9.09$

$T_{evap} = 20°F = T_1 \therefore h_1 = h_g = 106.383$ Btu/#m, $S_1 = S_g =$

.22379 Btu/#m°R

$T_{cond} = 80°F = T_3 \therefore P_3 =$ saturation pressure at $T_3 = 158$PSIA

$h_3 = h_f = 33.109$ Btu/#m $= h_4$

Assume isentropic compression, $S_1 = S_2$

$S_2 = .22379$ Btu/#m°R and $P_2 = P_3 = 158$PSIA imply that

$h_2 = 117.059$ Btu/#m

<u>Electric Cooling</u>: $COPW_c = \frac{\dot{Q}_{evap}}{\dot{W}} = \frac{\dot{m}(h_1-h_4)}{\dot{m}(h_2-h_1)} = \frac{(h_1-h_4)}{(h_2-h_1)}$

$$COPW_c = \frac{(106.383 - 33.109)\text{Btu/\#m}}{(117.059 - 106.383)\text{Btu/\#m}} = 6.86$$

Similarly, the other equations used are as follows:

Gas cooling: (assuming $\eta_{th} = .18$)

$$COPH_C = \frac{\eta_{th}\ \dot{Q}_{evap}}{\dot{W}} = \frac{\eta_{th}\ \dot{m}(h_1-h_4)}{\dot{m}(h_2-h_1)} = \frac{(.18)(h_1-h_4)}{(h_2-h_1)}$$

Electric heating:

$$COPW_H = \frac{\dot{Q}_{cond}}{\dot{W}} = \frac{\dot{m}(h_2-h_3)}{\dot{m}(h_2-h_1)} = \frac{(h_2-h_3)}{(h_2-h_1)}$$

Gas heating: (assuming $\eta_{th} = .18$, $F_{rec} = .60$)

$$COPH_H = \frac{\eta_{th}\ \dot{Q}_{cond}}{\dot{W}} + F_{rec}(1-\eta_{th}) = \frac{\eta_{th}\ \dot{m}(h_2-h_3)}{\dot{m}(h_2-h_1)} + F_{rec}(1-\eta_{th})$$

$$COPH_H = \frac{(.18)(h_2-h_3)}{(h_2-h_1)} + .49$$

These calculations were performed for nine sets of T_{evap} and T_{cond}.

In order to plot Figures 11, 12, 13, and 14, a relationship between the (T_L/T_H) parameter and COP was required. The following relations were formulated and solved for K_1 and K_2. The final values of K_1 and K_2 listed were, in each case, the average of the nine calculations performed.

<u>Electric cooling</u>: $COPW_C = K_1\left(\frac{T_L/T_H}{1-T_L/T_H}\right)$; $K_1 = .81$

<u>Gas cooling</u>: $COPH_C = K_1\left(\frac{T_L/T_H}{1-T_L/T_H}\right)$; $K_1 = .15$

<u>Electric heating</u>: $COPW_H = K_1\left(\frac{1}{1-T_L/T_H}\right)$; $K_1 = .83$

<u>Gas heating</u>: $COPH_H = K_1\left(\frac{1}{1-T_L/T_H}\right) + K_2$; $K_1 = .15$,

$K_2 = .49$

Similar calculations were performed to produce Figures 30, 31, 32, and 33. The only difference was the use of the actual experimentally measured cycle values instead of the ideal vapor-compression cycle values.

APPENDIX B

GAS FURNACE EXPERIMENTAL DATA

(The values tabulated below are the averages of three tests.) All temperatures are in °F.

	Low Heat Low Fan	Low Heat High Fan	High Heat Low Fan	High Heat High Fan
T9	152.3	138.3	191.3	165.0
T10	138.0	123.7	166.3	144.7
T11	123.3	108.7	149.0	126.0
T12	81.3	86.3	90.0	95.0
T13	83.0	88.0	93.3	94.7
T14	83.7	87.3	93.7	93.7
TE1	243.3	235.0	321.7	300.0
Water Flow Rate, (Gal/Min)	8.7	8.6	6.6	7.9
Gas Input, (Ft^3/Min)	1.2	1.2	1.5	1.5
Electric Input, (KW)	.37	.72	.41	.76

Sample η Calculation

For the case of low heat and low fan:

All ρ(density) values were taken from reference [18] and were considered to be the density at the average water temperature.

The heating value of the natural gas used was assumed to be 1,030 Btu/ft^3.

$$\eta = \frac{\dot{m}_w \, \Delta T_w \, C_{pw}}{\dot{Q}_{gas}} \tag{46}$$

$$\dot{m}_w = \dot{V}\rho = (8.65 \, \frac{gal}{min})(\frac{1 \, ft^3}{7.4805 \, gal})(\frac{61.29 \, \#m}{ft^3}) = 70.87 \, \frac{\#m}{min}$$

$$\Delta T_w = T_9 - T_{10} = (152.3 - 138.0)°F = 14.3°F$$

$$\dot{Q}_{gas} = \dot{V}(\text{heating value of gas}) = (\frac{1.2 \, ft^3}{min})(1,030 \, \frac{Btu}{ft^3}) =$$

$$1236.0 \, \frac{Btu}{min}$$

$$\eta = \frac{(70.87 \frac{\#m}{min})(14.3°F)(1 \, \frac{Btu}{\#m°F})}{1236.0 \, \frac{Btu}{min}} = .803$$

$$\eta = 80.3\%$$

APPENDIX C

ELECTRIC HEAT PUMP ANALYSIS

Electric Heat Pump Cooling Cycle Data

Eight tests were conducted. All temperatures are in °F and all pressures are in PSIG.

	Test #1	Test #2	Test #3	Test #4	Test #5	Test #6	Test #7	Test #8
T1	42	28	17	58	25	18	20	28
T2	190	180	168	216	216	186	192	216
T3	41	22	9	42	21	6	16	26
T4	50	30	16	51	31	15	23	29
T5	70	59	52	75	60	50	60	62
T6	48	26	15	52	30	16	19	29
T7	80	80	80	86	83	80	90	90
T8	192	180	165	215	216	188	189	211
T17	195	180	170	210	215	195	200	215
P1	57	40	28	62	34	28	30	40
P2	265	233	209	282	229	215	255	258
P3	252	230	205	280	226	206	250	252
P4	57	40	28	62	36	28	30	40
P5	80	58	39	85	58	40	45	55
P6	248	230	204	278	226	202	243	252
P7	252	230	205	280	228	202	250	252
Freon Flow Rate (gal/min)	1.18	0.66	0.56	1.18	0.62	0.50	0.56	0.60
Electric Input (KW)	7.85	6.32	5.76	8.79	6.32	5.76	5.89	6.65

The following thermocouples were installed to measure air temperatures at various points in and around the system: T11, T12, T13, T14, T15, and T16. Since they were installed primarily as a check and were not used in any of the calculations presented in this thesis, these values are not tabulated here.

Sample Calculations for Capacity and $COPW_c$

The calculations presented below were made for test #1. Thermodynamic properties were taken from reference [17]. Subcooled liquid properties were approximated by saturated liquid properties at that temperature. Enthalpy changes (Δh) were taken directly from a Pressure-Enthalpy Diagram (C-1) in reference [17].

$$T_7 = 80°\text{F}, \quad \text{thus} \quad V_f = .013492 \ \text{ft}^3/\#\text{m}$$

$$\dot{V}\left(\frac{\text{ft}^3}{\text{hr}}\right) = \left(1.18 \ \frac{\text{gal}}{\text{min}}\right)\left(60 \ \frac{\text{min}}{\text{hr}}\right)\left(\frac{1 \ \text{ft}^3}{7.4805 \ \text{gal}}\right) = 9.5 \ \frac{\text{ft}^3}{\text{hr}}$$

$$\dot{m}\left(\frac{\#\text{m}}{\text{hr}}\right) = \frac{\dot{V}}{V_f} = \frac{9.5 \ \text{ft}^3/\text{hr}}{.013492 \ \text{ft}^3/\#\text{m}} = 704.12 \ \frac{\#\text{m}}{\text{hr}}$$

$$\text{Capacity} \ \left(\frac{\text{Btu}}{\text{hr}}\right) = \dot{Q}_{evap} = \dot{m}(h_1 - h_4)$$

$$\text{Capacity} = \left(704.12 \ \frac{\#\text{m}}{\text{hr}}\right)(109-34)\frac{\text{Btu}}{\#\text{m}} = \left(5.28\text{x}10^4 \ \frac{\text{Btu}}{\text{hr}}\right)\left(\frac{\text{hr ton}}{12{,}000 \ \text{Btu}}\right)$$

$$= (4.40 \ \text{tons})$$

$$\dot{E}\left(\frac{Btu}{hr}\right) = (7.85\ kw)\left(\frac{56.89\ Btu}{min\ kw}\right)\left(\frac{60\ min}{hr}\right) = 2.68x10^4\ \frac{Btu}{hr}$$

$$COPW_C = \frac{\dot{Q}_{evap}}{\dot{E}} = \frac{5.28x10^4\ Btu/hr}{2.68x10^4\ Btu/hr} = (1.97)$$

Electric Heat Pump Heating Cycle Data

Four tests were conducted. All temperatures are in °F and all pressures are in PSIG.

	Test #1	Test #2	Test #3	Test #4
T1	74	73	56	50
T2	204	220	215	206
T3	201	214	208	201
T4	122	102	98	102
T5	116	97	90	90
T6	82	74	65	64
T7	74	69	58	48
T8	71	66	48	36
T17	220	244	238	226
P1	123	115	72	60
P2	345	370	342	314
P3	338	368	335	311
P4	337	366	333	310
P5	320	357	323	300
P6	129	121	96	81
P7	123	115	86	64
Freon Flow Rate, (gal/min)	1.62	1.58	1.40	1.03
Electric Input, (KW)	9.97	10.80	10.80	9.09

The following thermocouples were installed to measure air temperatures at various points in and around the system: T11, T12, T13, T14, T15, and T16. Since they were installed primarily as a check and were not used in any of the calculations presented in this thesis, these values are not tabulated here.

Sample Calculations for Capacity and $COPW_H$

The calculations presented below were made for test #1. Thermodynamic properties were taken from reference [17]. Subcooled liquid properties were approximated by saturated liquid properties at that temperature. Enthalpy changes (Δh) were taken directly from a Pressure-Enthalpy Diagram (C-1) in reference [17].

$$T_4 = 122°F, \quad \text{thus} \quad v_f = .014768\ ft^3/\#m$$

$$\dot{V}\left(\frac{ft^3}{hr}\right) = \left(1.62\ \frac{gal}{min}\right)\left(\frac{60\ min}{hr}\right)\left(\frac{1\ ft^3}{7.4805\ gal}\right) = 13.0\ \frac{ft^3}{hr}$$

$$\dot{m}\left(\frac{\#m}{hr}\right) = \frac{\dot{V}}{v_f} = \frac{13.0\ ft^3/hr}{.014768\ ft^3/\#m} = 880.96\ \frac{\#m}{hr}$$

$$\text{Capacity}\left(\frac{Btu}{hr}\right) = \dot{Q}_{cond} = \dot{m}(h_2 - h_3)$$

$$\text{Capacity} = \left(880.96\ \frac{\#m}{hr}\right)(132-45)\ \frac{Btu}{\#m} = \left(7.66x10^4\ \frac{Btu}{hr}\right)$$

$$\dot{E} = (9.97\ \text{kw})\left(\frac{56.89\ \text{Btu}}{\text{min kw}}\right)\left(\frac{60\ \text{min}}{\text{hr}}\right) = 2.68\text{x}10^{4}\ \frac{\text{Btu}}{\text{hr}}$$

$$COPW_{H} = \frac{\dot{Q}_{cond}}{\dot{E}} = \frac{7.66\text{x}10^{4}\ \text{Btu/hr}}{3.40\text{x}10^{4}\ \text{Btu/hr}} = (2.25)$$

APPENDIX D

GAS HEAT PUMP ANALYSIS

Gas Heat Pump Cooling Cycle Data

Three tests were conducted. All temperatures are in °F and all pressures are in PSIG.

	Test #1	Test #2	Test #3
T1	90	85	85
T2	192	186	190
T3	68	76	74
T4	41	52	56
T5	82	88	110
T6	47	54	62
T7	94	100	118
T8	187	183	187
TE1	172	170	165
TE2	1170	1175	1172
TE3	1758	1770	1760
P1	66	82	83
P3	211	235	297
P4	66	80	86
P5	71	86	93
P6	197	210	265
P7	208	231	292
Freon Flow Rate, (gal/min)	1.75	3.11	2.96
Natural Gas Flow Rate, (ft^3/min)	1.95	2.13	2.11
Electric Input, (KW)	1.85	1.85	1.85

The following thermocouples were installed to measure air temperatures at various points in and around the system: T11, T12, T13, T14, T15, and T16. Since they were installed primarily as a check and were not used in any of the calculations presented in this thesis, these values are not tabulated here. The exhaust temperatures were recorded, but the waste heat recovery calculation was not included in this case, of course, since the desired effect is cooling.

Sample Calculations for Capacity and $COPH_C$

The calculations presented below were made for test #1. Thermodynamic properties were taken from reference [17]. Subcooled liquid properties were approximated by saturated liquid properties at that temperature. Enthalpy changes (Δh) were taken directly from a Pressure-Enthalpy Diagram (C-1) in reference [17].

$$T_7 = 94°F, \quad \text{thus} \quad v_f = .013864\ ft^3/\#m$$

$$\dot{V}(\frac{ft^3}{hr}) = (1.75\ \frac{gal}{min})(\frac{60\ min}{hr})(\frac{1\ ft^3}{7.4805\ gal}) = 14.05\ \frac{ft^3}{hr}$$

$$\dot{m}(\frac{\#m}{hr}) = \frac{\dot{V}}{v_f} = \frac{14.05\ ft^3/hr}{.013864\ ft^3/\#m} = 1{,}013.50\ \frac{\#m}{hr}$$

$$\text{Capacity}\ (\frac{Btu}{hr}) = \dot{Q}_{evap} = \dot{m}(h_1 - h_4)$$

$$\text{Capacity} = (1{,}013.50\ \frac{\#m}{hr})(116.5-36)\frac{Btu}{\#m} = (8.16\text{x}10^4\ \frac{Btu}{hr})$$

$$(\frac{hr\ ton}{12{,}000\ Btu}) = 6.80\ \text{tons}$$

$$\dot{G}(\frac{Btu}{hr}) = \dot{V}(\frac{ft^3}{hr})(\text{heating value of natural gas, } Btu/ft^3)$$

Assume the heating value of methane to be 1,030 Btu/ft^3.

$$\dot{G} = (1.95\ \frac{ft^3}{min})(\frac{60\ min}{hr})(1{,}030\ \frac{Btu}{ft^3}) = 1.21\text{x}10^5\ \frac{Btu}{hr}$$

$$COPH_c = \frac{\dot{Q}_{evap}}{\dot{G}} = \frac{8.16\text{x}10^4\ Btu/hr}{1.21\text{x}10^5\ Btu/hr} = (.674)$$

Gas Heat Pump Heating Cycle Data

Four tests were conducted. All temperatures are in °F and all pressures are in PSIG.

	Test #1	Test #2	Test #3	Test #4
T1	104	106	102	92
T2	219	219	222	211
T3	183	190	186	186
T4	92	92	93	112
T5	88	86	85	104
T6	52	51	52	68
T7	68	65	66	58
T8	90	88	85	76
T9	114	113	115	124
T10	106	106	108	118
TE1	175	165	172	182
TE2	1190	1160	1175	1156
TE3	1680	1705	1705	1657
P1	103	93	94	85
P3	210	198	211	260
P4	206	196	206	255
P5	195	186	197	241
P6	115	110	113	98
P7	107	99	104	90
Water Flow Rate, (gal/min)	8.11	8.11	8.11	8.11
Freon Flow Rate, (gal/min)	1.62	1.40	1.58	1.71
Natural Gas Flow Rate, (ft^3/min)	1.85	1.68	1.71	2.03
Electric Input, (KW)	1.85	1.85	1.85	1.85

The following thermocouples were installed to measure air temperatures at various points in and around the system: T11, T12, T13, T14, T15, and T16. Since they were installed primarily as a check and were not used in any of the calculations presented in this thesis, these values are not tabulated here.

Sample Calculations for Capacity and $COPH_H$

The calculations presented below were made for test #1. Thermodynamic properties were taken from reference [17]. Subcooled liquid properties were approximated by saturated liquid properties at that temperature. Enthalpy changes (Δh) were taken from a Pressure-Enthalpy Diagram (C-1) in reference [17].

$$T_4 = 92°F, \quad \text{thus} \quad v_f = .013809 \ ft^3/\#m$$

$$\dot{V}(\frac{ft^3}{hr}) = (1.62 \ \frac{gal}{min})(\frac{60 \ min}{hr})(\frac{1 \ ft^3}{7.4805 \ gal}) = 13.0 \ \frac{ft^3}{hr}$$

$$\dot{m}(\frac{\#m}{hr}) = \frac{\dot{V}}{v_f} = \frac{13.0 \ ft^3/hr}{.013809 \ ft^3/\#m} = 942.14 \ \frac{\#m}{hr}$$

$$\text{Capacity} \ (\frac{Btu}{hr}) = \dot{Q}_{cond} = \dot{m}(h_2 - h_3)$$

$$\text{Capacity} = (942.14 \ \frac{\#m}{hr})(136-36) \ \frac{Btu}{\#m} = 9.42x10^4 \ \frac{Btu}{hr}$$

$$\dot{G}(\frac{Btu}{hr}) = \dot{V}(\frac{ft^3}{hr})(\text{heating value of natural gas, } \frac{Btu}{ft^3})$$

Assume the heating value of methane to be 1,030 $\frac{Btu}{ft^3}$.

$$\dot{G} = (1.85\ \frac{ft^3}{min})(\frac{60\ min}{hr})(1{,}030\ \frac{Btu}{ft^3}) = 1.14x10^5\ \frac{Btu}{hr}$$

$$COPH_H = \frac{\dot{Q}_{cond}}{\dot{G}} = \frac{9.42x10^4\ Btu/hr}{1.14x10^5\ Btu/hr} = .83 \quad \text{without waste heat recovery}$$

Sample Calculation of Waste Heat Recovery

Sample calculation of actual waste heat recovery:

$$\dot{Q}_{rec} = \dot{m}_w\ \Delta T_w\ C_{pw} \tag{47}$$

$$T_9 = 114°F, \quad T_{10} = 106°F, \quad \Delta T_w = T_9 - T_{10} = 8°F$$

The average temperature = 110°F which implies, from reference [18], that $\rho_w = 61.86\ \#m/ft^3$.

$$\dot{m} = \rho\dot{V} = (61.86\ \frac{\#m}{ft^3})(65.03\ \frac{ft^3}{hr}) = 4{,}022.76\ \frac{\#m}{hr}$$

$$\dot{Q}_{rec} = (4{,}022.76\ \frac{\#m}{hr})(8°F)(\frac{1\ Btu}{\#m\ °F}) = 32{,}182.08\ \frac{Btu}{hr}$$

$$\dot{Q}_{rec} = 32{,}182.08\ \frac{Btu}{hr}$$

For gas heat pump heating cycle test #1:

$$COPH_H = \frac{\dot{Q}_{cond} + \dot{Q}_{rec}(\text{actual})}{\dot{G}} = \frac{(9.42\text{x}10^4 + 32{,}182.08)\text{Btu/hr}}{1.14\text{x}10^5 \text{ Btu/hr}}$$

$$= 1.11$$

$COPH_H = 1.11$ with the actually measured waste heat recovery

<u>Sample Calculation of Possible Waste Heat Recovery</u>

The following calculation is based on the premise that the Wankel engine exhaust pipe can be insulated such that the heat loss will be minimized. The actual heat recovery is based on a temperature difference of (1175°F-170°F = 1005°F). Since the temperature of the exhaust gases at the combustion chamber outlet was measured to be as high as 1770°F, the following calculation gives an indication of the performance improvement possible with appropriate exhaust pipe insulation. The temperature difference considered is (1770°F-170°F = 1600°F). Assuming a stoichiometric methane combustion reaction,

$$CH_4 + 2O_2 + 2(3.76)N_2 \rightarrow CO_2 + 2H_2O + 7.52N_2$$

and using Table A.7 in reference [15] for the products of combustion,

$$CO_2: \quad C_{po}\left(\frac{Btu}{\#mole\,°R}\right) = 16.2 - \frac{6.53\times10^3}{T} + \frac{1.41\times10^6}{T^2}$$

$$H_2O: \quad C_{po}\left(\frac{Btu}{\#mole\,°R}\right) = 19.86 - \frac{597}{\sqrt{T}} + \frac{7500}{T}$$

$$N_2: \quad C_{po}\left(\frac{Btu}{\#mole\,°R}\right) = 9.47 - \frac{3.47\times10^3}{T} + \frac{1.16\times10^6}{T^2}$$

a comparison between the actual heat recovery and the heat recovery that ideally could be achieved with an insulated exhaust can be performed.

$$CO_2: \; dh = \int_{T=630°R}^{T=2230°R} Cp\, dT = \left(1.93\times10^4 \frac{Btu}{\#mole}\right)\left(\frac{44\#mole}{\#m}\right) = 8.49\times10^5 \frac{Btu}{\#m}$$

$$CO_2: \; dh = \int_{T=630°R}^{T=1635°R} Cp\, dT = \left(1.14\times10^4 \frac{Btu}{\#mole}\right)\left(\frac{44\#mole}{\#m}\right) = 5.02\times10^5 \frac{Btu}{\#m}$$

$$H_2O: \; dh = \int_{T=630°R}^{T=2230°R} Cp\, dT = \left(1.49\times10^4 \frac{Btu}{\#mole}\right)\left(\frac{18\#mole}{\#m}\right) = 2.68\times10^5 \frac{Btu}{\#m}$$

$$H_2O: \; dh = \int_{T=630°R}^{T=1635°R} Cp\, dT = \left(8.90\times10^3 \frac{Btu}{\#mole}\right)\left(\frac{18\#mole}{\#m}\right) = 1.60\times10^5 \frac{Btu}{\#m}$$

$$N_2: \; dh = \int_{T=630°R}^{T=2230°R} Cp\, dT = \left(1.21\times10^4 \frac{Btu}{\#mole}\right)\left(\frac{28\#mole}{\#m}\right) = 3.39\times10^5 \frac{Btu}{\#m}$$

$$N_2: \quad dh = \int_{T=630°R}^{T=1635°R} Cp\ dT = (7.39x10^3 \frac{Btu}{\#mole})(\frac{28\#mole}{\#m}) = 2.07x10^5 \frac{Btu}{\#m}$$

The enthalpy change of the mixture was calculated as follows:

$$dh_{mixture} = \frac{\sum_{i=1}^{3} dh_i \ M_i}{M_{mixture}} \qquad (50)$$

where:

M = mass

i refers to the ith component

$$dh_{\substack{mixture \\ (2230°R)}} = \frac{(8.49x10^5 \frac{Btu}{\#m})(1 \text{ mole } CO_2)(\frac{44\#m}{mole}) + (2.68x10^5 \frac{Btu}{\#m})(2 \text{ moles } H_2O)(\frac{18\#m}{mole}) + (3.39x10^5 \frac{Btu}{\#m})(7.52 moles N_2)(\frac{28\#m}{mole})}{(1 mole CO_2)(\frac{44\#m}{mole}) + (2 moles H_2O)(\frac{18\#m}{mole}) + (7.52 moles N_2)(\frac{28\#m}{mole})}$$

$$dh_{\substack{mixture \\ (2230°R)}} = 4.07x10^5 \frac{Btu}{\#m}$$

$$\underset{(1635°R)}{dh_{mixture}} = \frac{(5.02x10^{5}\frac{Btu}{\#m})(1\ mole\ CO_2)(\frac{44\#m}{mole}) + (1.60x10^{5}\frac{Btu}{\#m})(2moles H_2O)(\frac{18\#m}{mole}) + (2.07x10^{5}\frac{Btu}{\#m})(7.52moles N_2)(\frac{28\#m}{mole})}{(1mole CO_2)(\frac{44\#m}{mole}) + (2moles H_2O)(\frac{18\#m}{mole}) + (7.52moles N_2)(\frac{28\#m}{mole})}$$

$$\underset{(1635°R)}{dh_{mixture}} = 2.46x10^{5}\ \frac{Btu}{\#m}$$

$$\frac{\dot{Q}_{rec}(\text{insulated exhaust})}{\dot{Q}_{rec}(\text{actual})} = \frac{\eta_{ex}\dot{m}_{ex} \int_{T=630°R}^{T=2230°R} Cp\ dT}{\eta_{ex}\dot{m}_{ex} \int_{T=630°R}^{T=1635°R} Cp\ dT} = \frac{dh_{mixture(2230°R)}}{dh_{mixture(1635°R)}} = 1.65$$

$$\dot{Q}_{rec}(\text{insulated exhaust}) = 1.65\ \dot{Q}_{rec}(\text{actual})$$

$$\dot{Q}_{rec}(\text{actual}) = 32{,}182.08\ \text{Btu/hr}$$

$$\dot{Q}_{rec}(\text{insulated exhaust}) = 53{,}100.43\ \text{Btu/hr}$$

For gas heat pump heating cycle test #1:

$$COPH_H = \frac{\dot{Q}_{cond} + \dot{Q}_{rec}(\text{insulated exhaust})}{\dot{G}}$$

$$= \frac{(9.42x10^4 + 53{,}100.43)\text{Btu/hr}}{1.14x10^5\ \text{Btu/hr}} = 1.29$$

$COPH_H = 1.29$ with the ideal insulated exhaust waste heat recovery

APPENDIX E

SAMPLE CALCULATIONS FOR FIGURES 35 AND 36

Electric cooling: From Figure 30 at $T_L = 40°F = 500°R$ and $T_H = 120°F = 580°R$ or $\frac{T_L/T_H}{1-T_L/T_H} = 6.14$, $COPW_c = 1.90$ using a cost factor of \$.03/KW-hr,

$$\left(\frac{1}{1.90}\right)\left(\frac{\$.03}{\text{kw-hr}}\right)\left(\frac{\text{kw-hr}}{3413.40\text{Btu}}\right)\left(\frac{10^6}{10^6}\right) = \left(\frac{4.63}{10^6\text{Btu}}\right)$$

or taking the reciprocal:

$$\frac{10^6\text{Btu}}{\$4.63} = \left(\frac{2.16\text{x}10^5\text{Btu}}{\$}\right)$$

Gas heating (without waste heat recovery): From Figure 33 at $T_L = 40°F = 500°R$ and $T_H = 120°F = 580°R$ or $\frac{1}{1-T_L/T_H} =$ 7.14, $COPH_H = .57$

using a cost factor of $\$1.30/10^6$ Btu,

$$\left(\frac{1}{.57}\right)\left(\frac{\$1.30}{10^6\text{Btu}}\right) = \left(\frac{\$2.28}{10^6\text{Btu}}\right)$$

or taking the reciprocal:

$$\frac{10^6\text{Btu}}{\$2.28} = \left(\frac{4.39\text{x}10^5\text{Btu}}{\$}\right)$$

BIBLIOGRAPHY

1. A Study of Natural Gas Rotary Engine Driven Heat Pumps, First Annual Report, Georgia Institute of Technology, S. V. Shelton (principal investigator), July, 1974.

2. Kennedy, D., "The Potential of the Heat Activated Heat Pump," American Gas Association Monthly, April, 1973.

3. Stoecker, W. F., Refrigeration and Air Conditioning, McGraw-Hill Book Co., 1958.

4. Amana Electric Gas Package Unit, Service Instructions, Amana Refrigeration, Inc., Amana, Iowa.

5. Installation, Operation, and Maintenance Instructions for Electric-Gas Package Unit Cooling and Heating, Part No. A46815-4, Amana Refrigeration, Inc., Amana, Iowa.

6. Calvert, F. O., Harden, D. G., "A Comparative Study of Residential Energy Usage," Proceedings of the Intersociety Energy Conversion Engineering Conference, Paper 739103 (1973) p. 410-415.

7. Schweitzer, Gerald, Ebeling, A., Basic Air Conditioning, Volume 1, Hayden Book Co., Inc., New York, 1971.

8. Linzmaier, J., Personal Correspondence, Tecumseh Products Company, Atlanta, Georgia, 30340, April, 1975.

9. Ranco Reversing Valves, Bulletin 1919-1, Ranco, Inc., 601 West Fifth Avenue, Columbus 1, Ohio.

10. Granet, Irving, Thermodynamics and Heat Power, Reston Publishing Co., Inc., Reston, Virginia, 1974.

11. Perry, H. Allen, "On the Way: A Completely Different Auto Engine," The American Legion Magazine, March, 1972, p. 18-23, 42-45.

12. Norbye, Jan P., The Wankel Engine, Chilton Book Co., Philadelphia, Pennsylvania, 1971.

13. Keller, Helmut, "Small Wankel Engines," Society of Automotive Engineers, Paper No. 680572, 1968.

14. Trane Model G Compressors, Catalog DS-COM3, The Trane Company, La Crosse, Wisconsin, 54601, 1970.

15. Van Wylen, Gordon J., Thermodynamics, John Wiley & Sons, Inc., 1959.

16. The Wall Street Journal, August 7, 1974, page 8.

17. Thermodynamic Properties of Freon 22 Refrigerant, Booklet T-22, E. I. DuPont de Nemours & Co., Inc., Wilmington, Delaware 19898, 1964.

18. Streeter, Victor L., Fluid Mechanics, McGraw-Hill Book Company, 1958.